POWER

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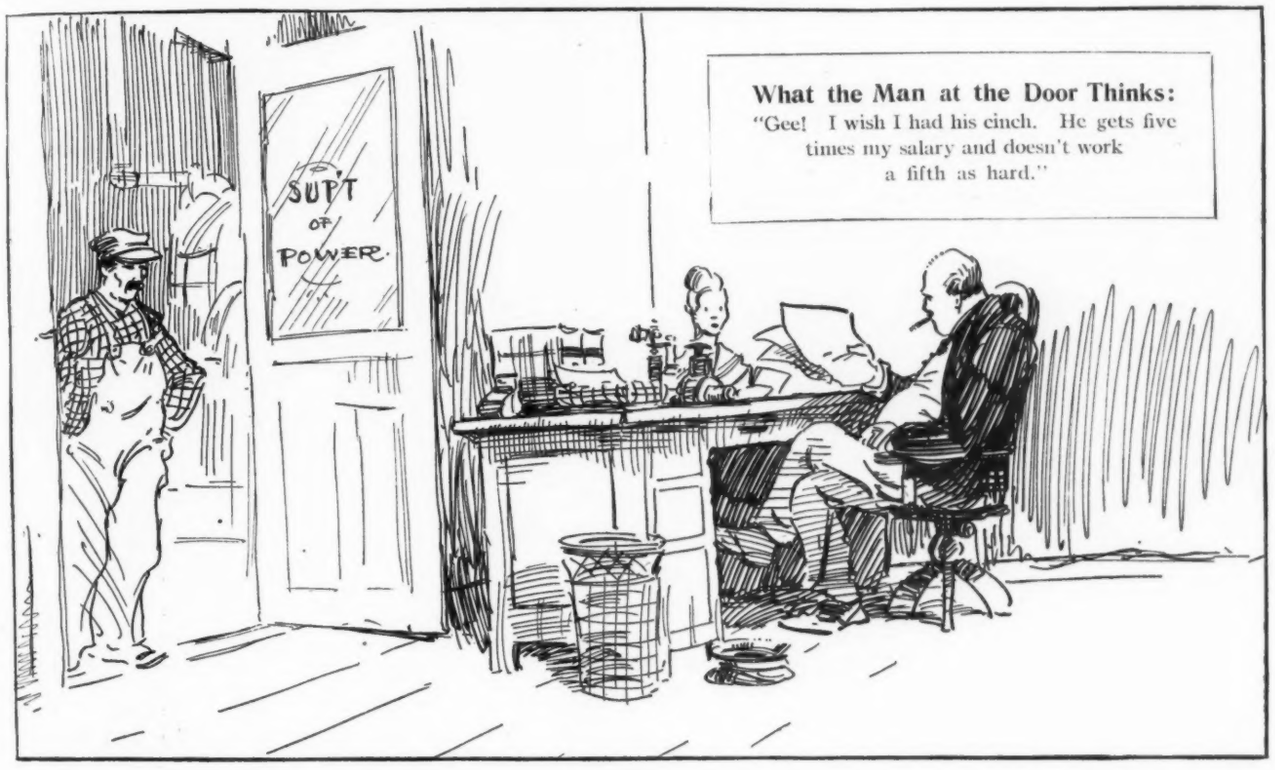
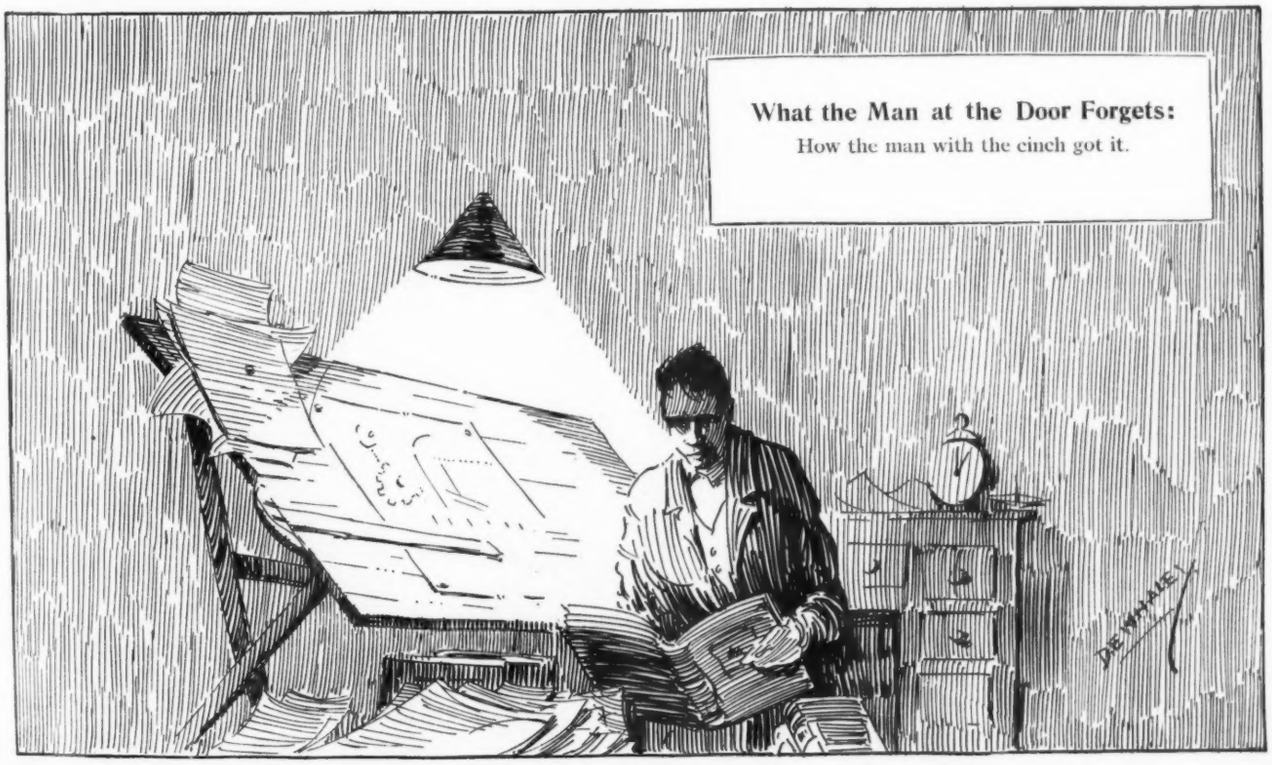


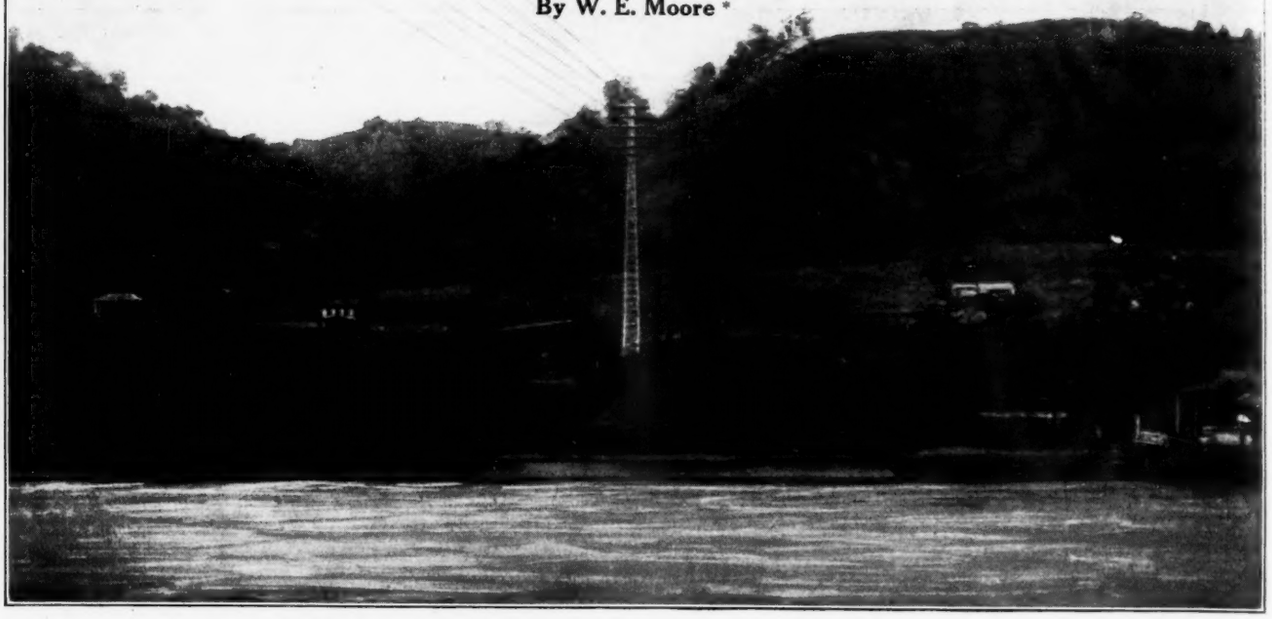
What the Man at the Door Thinks:
"Gee! I wish I had his cinch. He gets five times my salary and doesn't work a fifth as hard."



What the Man at the Door Forgets:
How the man with the cinch got it.

West Penn Traction Co.'s Wheeling Power Plant

By W. E. Moore *



SYNOPSIS—This is the second largest power plant of the West Penn Traction Co. It has been remodeled and contains turbine- and engine-driven units. One new boiler has been added to the six water-tube boilers, which have been reset. Forced draft is used, the fans being driven by steam turbines that can be operated at three speeds, ranging from 1,200 to 2,400 r.p.m. An ingenious idea has been applied in the handling of coal to and from the crusher. River water is used, and owing to its acidity, a water-softening system with an hourly capacity of 16,000 gal. has been installed.

The City of Wheeling, W. Va., lies on the western boundary of the territory served by the West Penn Traction Co., and here the company has in service its second largest power plant. The power-house site is on the Ohio River, which provides ample circulating water. The river is the only source of abundant water supply in that section and is contaminated by acid and scale-forming matter. Another disadvantage is the wide fluctuation in the river level from season to season. While the water is not altogether satisfactory for surface condensers, on account of the suspended matter found, the advantages to be derived from this type of equipment are evident in view of variation in the water level.

The rapidly increasing use of central-station power in this district during the last year or two has necessitated much additional power-plant equipment.

The equipment in the Forty-Second Street power house of the Wheeling Traction Co., a subsidiary of the West Penn Traction Co., originally consisted of two 650-kw. d.-c. engine generators, one 800-kw. engine generator and one 1,875-kv.-a. low-pressure steam turbine taking the exhaust from the engines. The low-pressure turbine is served by a jet condenser, the cooling water for which is pumped from a cold well (Fig. 2) at the river side. Fig. 1 is a general view of the engine room from the end containing the low-pressure turbine.

The company owned considerable land adjacent to the power house—sufficient to allow of building an 80-ft. extension. To take care of the increasing load, two of the 1,500-kw. maximum-rated turbines were removed from the Connellsville station and installed in the Wheeling plant, Fig. 3. A plan of the completed plant is shown in Fig. 4.

The old boiler room and a part of the engine-room floor level were below the high-water mark. The boilers were also too crowded for good operating conditions, particularly regarding cleaning and inspection. As the old settings were in unsatisfactory condition, the boilers were reset on concrete foundations and elevated sufficiently to bring the boiler-room floor above the high-water level. This also permitted of the laying of tracks of standard street-railway gage and of cars being run under the ashpits, thus providing for convenient removal of the ashes.

A list of apparatus installed in the improved plant is given in the table of principal equipment.

The boiler equipment of the old plant consisted of three 550-hp. and three 350-hp. special five-drum water-tube boilers, all of the same cross-section. These boilers were designed for 160 lb. steam pressure. The changes made consisted in adding one 550-hp. unit. The existing boilers were then reset and relocated. The new boiler was first erected on its foundation at the end of the new extension farthest removed from the old boiler room. The foundation for the second boiler was then prepared. After the concrete was set sufficiently, the boiler in the old boiler room next the extension was torn down and placed upon the new foundation. This left a clear working space be-

*Vice-president and general manager of the West Penn Traction Co.

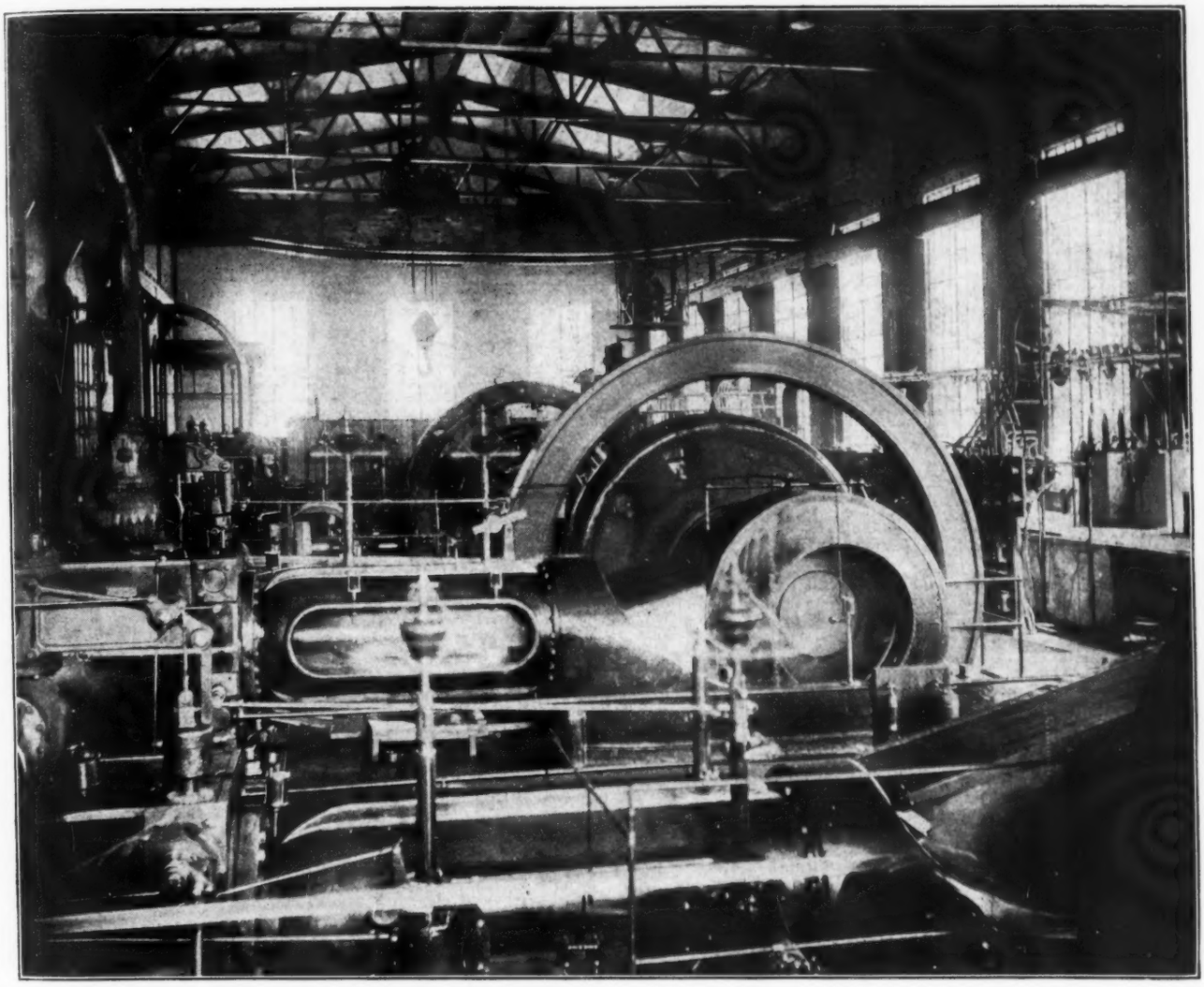


FIG. 1. GENERAL VIEW OF THE ENGINE ROOM FROM THE LOW-PRESSURE TURBINE END

tween the completed new part of the plant and the old part still in operation. The boilers are equipped throughout with built-in soot blowers.

Under the 550-hp. boilers are mechanical underfeed stokers with a grate area of 91 sq.ft., and in the 350-hp. boilers, stokers of the same kind having a grate area of 63 sq.ft. Fig. 5 is a view of part of the boiler room after the alterations. The conveyor housing is at the left.

The forced-draft equipment consists of seven turbo-conoidal fans, one supplying each boiler. One reëntry turbine is provided for each two fans. This arrangement is shown in Fig. 6. The speed of the unit varies from 1,200 to 2,400 r.p.m., according to the demand for air in the boiler furnace. The turbines are equipped with two nozzles, one of which has almost twice the capacity of the other. This gives three different operating speeds, the

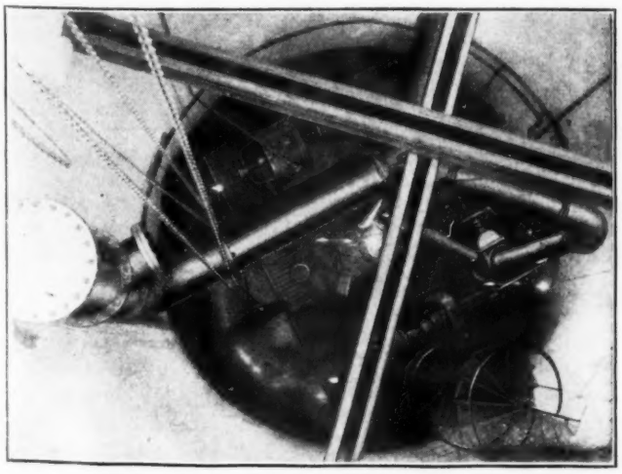


FIG 2. LOOKING INTO THE PUMP WELL, 48 FT. DEEP

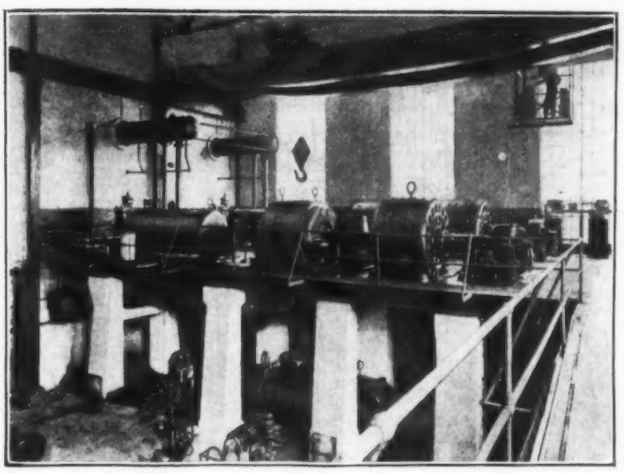


FIG. 3. HIGH-PRESSURE TURBO-GENERATOR UNITS

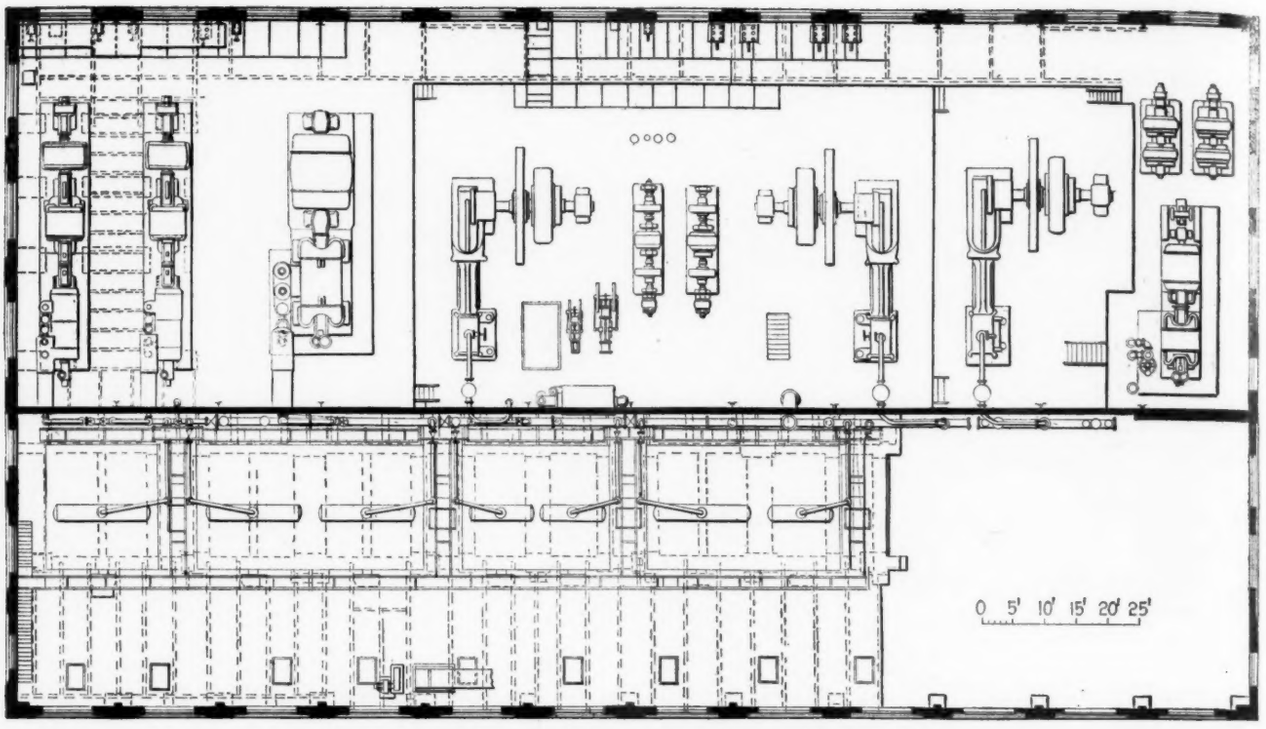


FIG. 4. PLAN VIEW OF THE WHEELING POWER PLANT

highest being obtained with both nozzles wide open, intermediate speeds being obtained by employing nozzles singly. The control of the air is by hand, the steam pressure to each turbine being regulated by the nozzle valves on the turbine. The steam supply for the stokers and fans is taken from the same header. The fans are in the basement under the firing alley in front of the boilers. This alley is about 22 ft. wide and gives ample room in which to repair the fans and turbines. The forced-draft equipment is liberally designed and when operated at its maximum capacity will force the boilers to 300 per cent. rating. It is not intended that the boilers will ordinarily be operated at over 200 per cent. rating.

The plant is laid out for an ultimate installation of three steel stacks, each of which will take care of about 1,600 boiler-horsepower. The two stacks now in place are each about 8½ ft. in diameter and 140 ft. above the boiler-room floor.

Cheap coal is brought into the plant on a spur from the Pennsylvania Lines from near-by mines. The coal-handling apparatus is shown in Fig. 7, a section through the plant which also outlines the elevator and conveyor design. The coal is dumped from drop-bottom cars into a concrete hopper and delivered to a scraper conveyor by a reciprocating feeder. The scraper conveyor carries all lump and large-sized coal to a single-roll crusher. The slack is sifted by the scraper conveyor in passing over a stationary screen.

An ingenious idea has been applied in the handling of the coal to and from the crusher. The scraper conveyor first passes over the crusher in delivering the lump coal and then by suitable sprocket wheels the conveyor travels back under the crusher and discharges the coal into a chute that directs the crushed coal upon the elevating bucket conveyor.

The bucket elevator is inclined so that the discharge end will be at the middle of the boiler room. After being elevated the coal is distributed by a shuttle conveyor driven by a 3-hp. 650-volt d.-c. motor; the longitudinal movement of the shuttle is by hand and controlled by a conveniently located hand-winch. The coal is crushed to such fineness as will pass through a 1¼-in. screen.

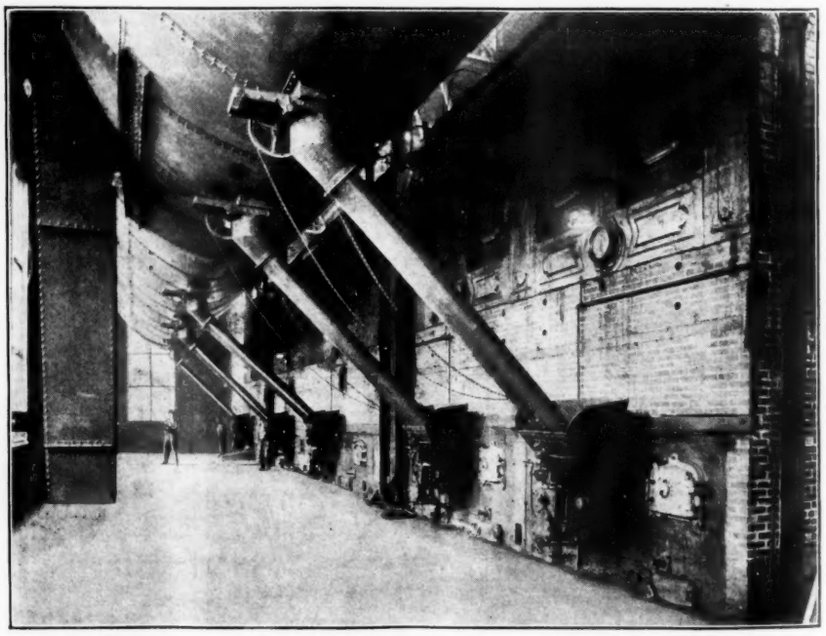


FIG. 5. VIEW IN FRONT OF THE BOILERS, CONVEYOR CASING AT THE LEFT

Less than 1 in. is sifted out through a flat screen, relieving the crusher and thus increasing its effective capacity. The coal is stored in an unlined plate-steel overhead bunker. The maximum amount of coal to be stored per lineal foot is about nine tons. The bunker is sealed by a partition wall, to keep dust from the boiler room. The capacity of the coal-handling apparatus, which is motor driven, is 75 tons per hour.

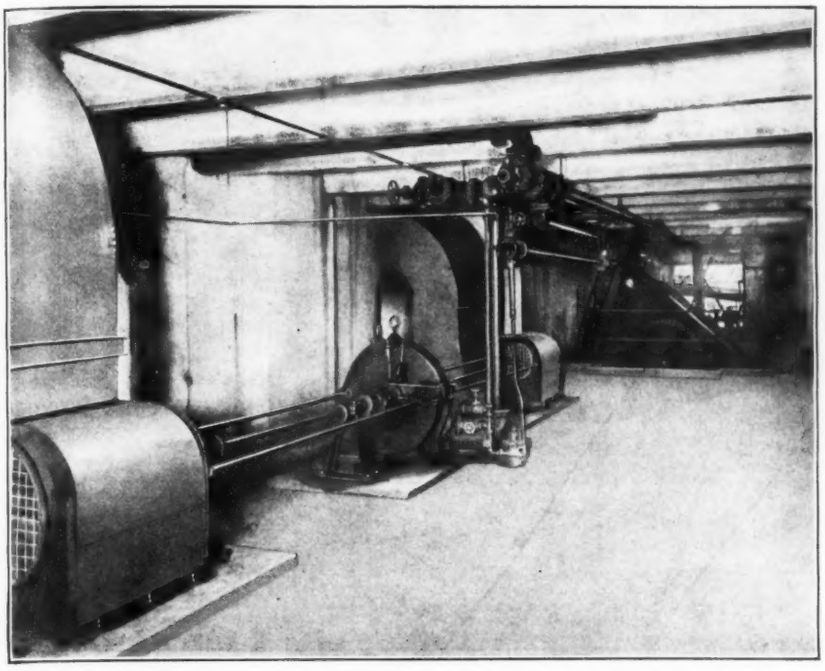


FIG. 6. THREE-SPEED TURBINE DRIVING FORCED-DRAFT FANS

The boilers are set over deep concrete ashpits having rolling doors on the bottom. The ashes are removed by the Wheeling Traction Co. in bottom-dump cars and are used for ballast on the car lines. These ash cars are switched immediately under the pits and the ashes dumped into them.

Two units only are at present equipped with surface condensers. Since the exhaust from the engines is consumed in the low-pressure turbine, with which a jet condenser is connected, this condensate is lost. A large amount of makeup water is therefore required, and on account of the unsatisfactory quality of the river water, the liability of much trouble with the boilers, from corrosion and the formation of scale, continually confronts the operators. At seasons of low water when mills upstream are working at full capacity, it is difficult to keep the boilers in service owing to the increase of acid in the water. To cope with the problem, a water-softening system of the "intermittent" type, having an hourly capacity of 16,000 gal., was installed. It is housed in a brick-concrete building, the filtered water discharging into a large concrete clear-well under the building. An interior view of the plant with the fillers in the foreground is shown in Fig. 8.

The softening plant has successfully neutralized the river water and has also removed all of the scale-forming salts and other matter. After the boilers had been op-

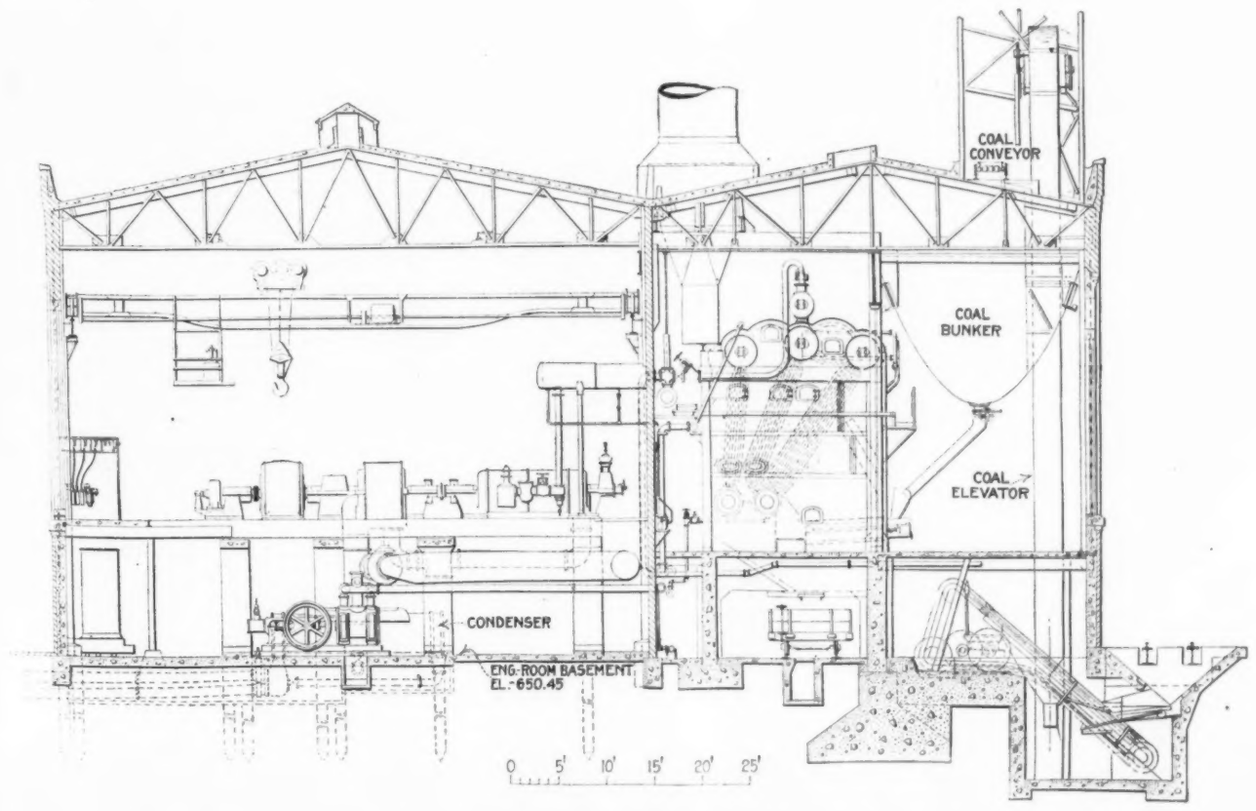


FIG. 7. SECTIONAL ELEVATION THROUGH BOILER AND TURBINE ROOMS AND BASEMENTS

erated three months, they were opened, and found free from scale. The water is pumped from the clear-well to a tank on the roof by two reciprocating steam pumps, either of which has sufficient capacity for the service. From Fig. 9, the flexibility of the water-supply system will be apparent. The condensate from the turbine condensers is pumped to the heater and the makeup-boiler supply is drawn from the tank on the roof, the water in the heaters being maintained at a constant level. The gland-water supply and all house service, including cooling water for transformers, are taken from this same source. The two boiler-feed pumps are of the outside packed, plunger type, which were part of the equipment of the old plant.

The plant has automatic boiler-feed regulators with bypass of the constant-flow type.

Turbine and Engine-Room Equipment

The turbine units are on concrete bases at a height of about 14 ft. above the basement floor and about 4 ft. above the maximum high-water level. The walls of the engine room are waterproofed so that the plant can run at all stages of the river. Cooling water for the condensers is pumped by two 12-in. centrifugal pumps in the bottom of the dry-well, Fig. 2, the inside diameter of which is 25 ft. and the depth 18 ft. The water is pumped into a common pressure pipe and distributed to the various condensers. The discharge from the condensers is collected in a common discharge pipe, the end

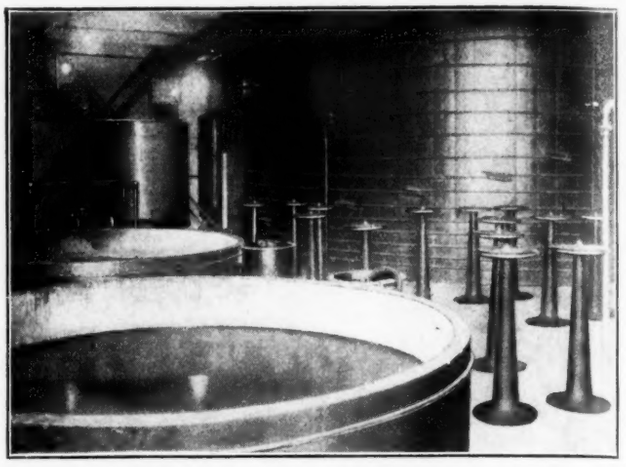


FIG. 8. PARTIAL VIEW OF THE WATER-SOFTENING PLANT

The electrical installation consists of two 1,500-kw. steam-turbine-driven generators, Fig. 3; one 1,875-kw. low-pressure turbo-generator; two 500-kw., 600-volt, d.-c. synchronous motor-generator sets; four 250-volt, 150-kw. railway boosters; two 600-kw., direct-connected railway generators, and one 850-kw., direct-connected railway generator. All of the turbo-generators and motors are star-connected, and the turbo-generators have the neutral grounded.

The 2,200-volt busbars are mounted on pipe framework and inclosed in "transite" cells. All the switches are remote-controlled, electrically operated from the exciter system. A system of busbar disconnecting switches has been installed so that the plant may be operated as one unit in parallel with the high-tension system, or as two separate units, one unit carrying the railway system and

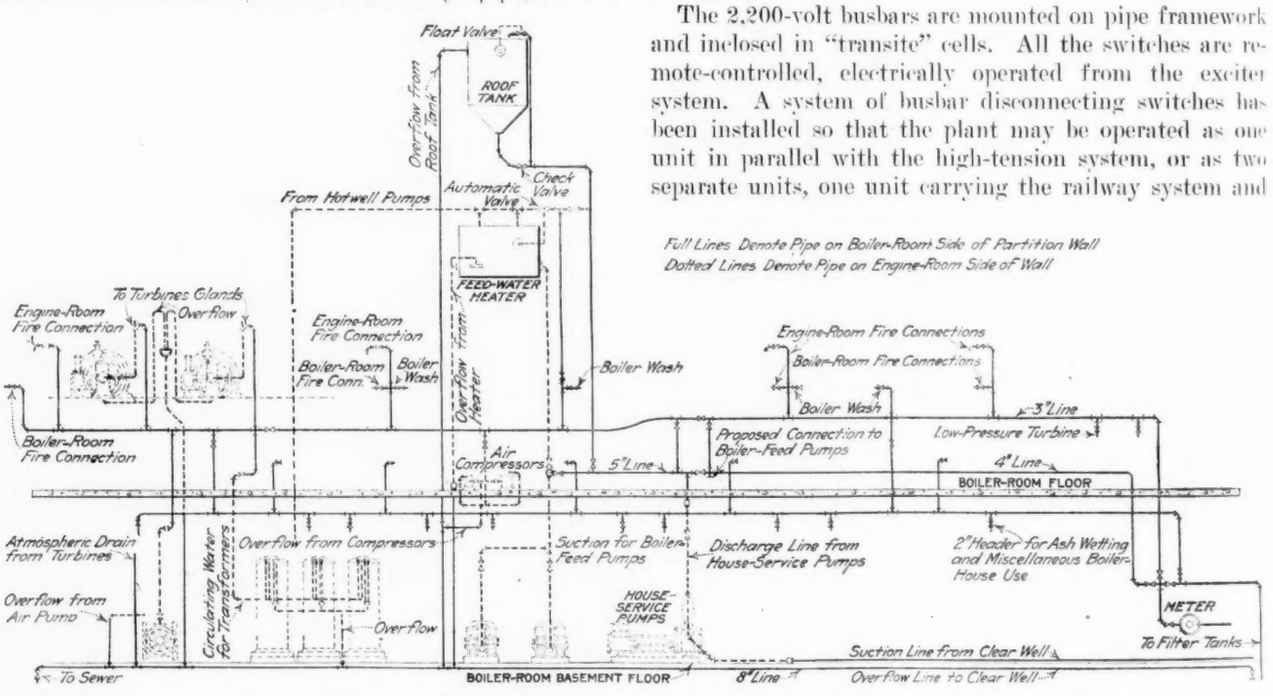


FIG. 9. LAYOUT OF THE WATER-SUPPLY SYSTEM, SHOWING ITS FLEXIBILITY

of which is sealed at a little above pool stage, thus establishing a circulating "loop," and so reducing the working head on the pumps. From the river water at pool stage (when the wickets in the dams along the river are raised during the dry seasons) to the top of the condensers is 42 ft.

It is the intention to operate the engines on the railway load, the three turbo-generators being alternating current. The power generated by these units will be transformed to 22,000 volts and delivered to the company's transmission system.

The 22,000-volt busses are mounted on pipe framework above the automatic oil switches. Disconnecting switches have been installed for sectionalizing these busses, and a special method has been arranged for jumping and disconnecting oil switches for inspection or repairs, without interrupting the service. Low equivalent light-

ning arresters are on incoming and outgoing high-tension lines.

The exciter system is supplied by direct-connected exciters on each of the turbines. These exciters are operated in parallel in connection with the regulator which maintains the station voltage. The exciter busses are near the 2,200-volt busses, and all switches and rheostats are remote, electrically controlled from the switchboard. A voltage regulator on the motor-generator exciter system supplies power-factor correction from the motor-generator sets, all of which are designed for 80 per cent. power factor at full load.

The railway boosters consist of two 150-kw. 250-volt d.-c. generators, directly connected to one 435-kw. synchronous motor. These generators have shunt and series fields so that they may be operated either as two independent railway boosters or as one 150-kw. 500-volt railway generator. Switching arrangements allow for changing from boosters to generators. These machines are usually operated as boosters during the heavy railway loads and as motor-generator sets on the lighter loads.

There are three 2,000-kw. water-cooled transformers, connected star on the low-tension side and delta on the high-tension side. They are standard 2,310-23,100-volt

PRINCIPAL EQUIPMENT OF WEST PENN COMPANY POWER HOUSE, WHEELING, W. VA.

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
2	Turbo-generators	Single, double-flow, separate casing	1,500-kw	Main generators	1,500 r.p.m., 150 lb. steam, 2,700 volts, 3-phase, 60-cycle	Westinghouse Mach. Co. / Westinghouse Elec. & Mfg. Co.
2	Engines	Corliss	32x48-in	Driving railway generators	150 lb. steam, non-condensing, 90 r.p.m.	C. & G. Cooper Co.
2	Generators	D.-c. engine-type	600-kw	Railway service, engine driven	90 r.p.m., 1,000 amp., 550-600 volts	Bullock Electric Co.
1	Engine	Reciprocating Corliss	34x60-in	Driving railway generator	150 lb. steam, non-condensing, 75 r.p.m.	C. & G. Cooper Co.
1	Generator	D.-c., engine-type	800-kw	Railway service, engine driven	75 r.p.m., 1,322 amp., 605 volts	Westinghouse Elec. & Mfg. Co.
1	Turbo-generator and exciter	Mixed-pressure turbine a.-c. and d.-c. gen.	1,875-kv.-a., 37.5 kw	Main unit	Exhaust steam, 1,800 r.p.m., 2,400 volts, 3-phase, 60-cycles, exciter d.-c. gen. 125 volts	Westinghouse Machine Co. / Westinghouse Elec. & Mfg. Co.
2	Condensers	Surface	4,000 sq.ft.	With 1,500-kw. turbines		Alberger Pump & Condenser Co.
1	Condenser	Leblanc	No. 17	With low-pressure turbine		Westinghouse Machine Co.
1	Turbine	Single runner		Driving Leblanc condenser pumps	1,000 r.p.m., 150 lb. steam	Westinghouse Machine Co.
1	Pump	Mullan, air	10x26x14-in	With surface condensers	100 r.p.m.	C. H. Wheeler Mfg. Co.
2	Motor-generators	Ry. boosters and motor-gen. sets	Motors 425 hp., gen. 150-kw.	Each motor drives two ry. boosters. Can be operated as two boosters or as one generator	2,300 volts, 113 amp., 600 r.p.m., 3-phase, 60-cycle, d.-c. 250 volts, 600 amp.	Allis-Chalmers Co.
2	Motor-generator	A.-c. motor, d.-c. generator	720-hp	Railway service	2200 volts, 270 r.p.m., 3-phase, 60-cycle, d.-c. 600 volts, exciter 125 volts	Westinghouse Elec. & Mfg. Co.
7	Boilers	Stirling, water-tube, five drum	4-550-hp., 3-350-hp.	Steam generators	150-lb. pressure	Babcock & Wilcox Co.
7	Stokers	Underfeed	63 sq.ft.	Boiler furnaces	Forced-draft	Combustion Engineering Corporation
7	Fans, blowers	Conoidal	No. 4	For forced draft to furnaces	1,200 to 2,400 r.p.m.	Buffalo Forge Co.
5	Turbines	Three-speed		Driving forced-draft fans	150 lb. steam, 1 lb. back-pressure, 1,200-2,400 r.p.m.	Westinghouse Machine Co.
1	Heater	Feed water		Heating boiler feed water	Exhaust steam from auxiliaries	Warren Webster Co.
1	Crane	Travelling	30-ton	In turbine room	5 hp. motor, carriage, one bridge motor	Crane, Cleveland Crane & Car Co. / Motors, Westinghouse Elec. & Mfg. Co.
2	Pumps	Circulating	12 in., 4,000 gal. per min.	In pump pit, station supply	Motor-driven, 50-ft. head, 915 r.p.m.	Allis-Chalmers Co.
2	Motors	Direct-current	75-hp	Driving circ. pumps	900 r.p.m., 600 volts	Allis-Chalmers Co.
1	Pump	Vertical, centrifugal	500 gal. per min.	With filtered-water supply	Motor-driven, 1,155 r.p.m.	Reinecke-Wagner Pump & Supply Co.
1	Motor	Vertical type	15-hp	Driving vertical centrifugal pump	1,155 r.p.m., 220 volts	Westinghouse Elec. & Mfg. Co.
1	Starter	Automatic		With 15-hp. motor	220 volts	Westinghouse Elec. & Mfg. Co.
1	Motor-pump	D.-c. and centrifugal	¾-hp	Softening solution	650 volts, 1.35 amp., 725 r.p.m.	Motor, General Electric Co. / Pump, Henry R. Worthington
2	Pumps	Duplex, center packed	14x8½x18-in	Boiler-feed	Variable-speed	Snow Steam Pump Co.
2	Pumps	Simplex	10x9x12-in	House service	Variable-speed	Union Steam Pump Co.
2	Pumps	Duplex	6x5½x6-in	Hotwell		Alberger Pump & Condenser Co.
1	Crusher	Single roll	24x30-in	Crushing coal	75 tons per hr.	Pennsylvania Crusher Co.
1	Motor	Direct-current	30-hp	Driving coal crusher	Intermittent, 650 volts, 975 r.p.m.	Westinghouse Elec. & Mfg. Co.
1	Conveyor	Scraper	30x12-in., 75-ton per hr.	Coal to crusher	Intermittent, motor-driven	Dempcy-Degener Co.
1	Conveyor	Bucket	20x11¼x10-in., 75-ton per hr.	Coal to belt conveyor	Intermittent, motor-driven	Dempcy-Degener Co.
1	Conveyor	Shuttle, belt	18-in., 75 ton per hr.	Coal to bunker	Intermittent, motor-driven	Dempcy-Degener Co.
1	Motor	Direct-current	15-hp	Drives bucket conveyor	Intermittent, 650 volts, 20.4 amp., 825 r.p.m.	Westinghouse Elec. & Mfg. Co.
1	Motor	Direct-current	5-hp	Drives shuttle conveyor	Intermittent, 650 volts, 7 amp., 1,200 r.p.m.	Allis-Chalmers Mfg. Co.
1	Compressor	Air	10x10x10-in	Station use	Intermittent	Ingersoll-Sargent Drill Co.
1	Compressor	Air	12x12x14-in	Station use	Intermittent	Stillwell, Bierce & Smithvale Co.
1	Softener	Water	16,000-gal	Softening boiler-feed water	Intermittent	Pittsburgh Filter Mfg. Co.
1	Compressor	Air	4¾x4¾-in	Agitating water to be filtered	Intermittent, motor-driven	Clayton Air Co.
1	Motor	Direct-current	3-hp	Drives compressor	1,575 r.p.m., 650 volts, 4.15 amp.	General Electric Co.
3	Transformers	Water cooled	2,000-kv.-a.	Main step-up	2,200-22,000 volts, single-phase	Allis-Chalmers Mfg. Co.
1	Transformer		15-kv.-a.	House lighting	2,200 volts to 220-110 volts	Westinghouse Elec. & Mfg. Co.
3	Regulators	Induction	22-kw.	Line, regulation	2,200 volts primary, 100 volts secondary, amps. 10% regulation	Westinghouse Elec. & Mfg. Co.
3	Motors	Alternating current		With induction regulators	1,200 r.p.m., 220 volts, single-phase, 6-cycle	Westinghouse Elec. & Mfg. Co.
3	Relays			On regulators	110 volt, 60-cycle	Westinghouse Elec. & Mfg. Co.
2	Switches	Oil circuit-breakers			Electrically operated, 2,200 volts, 900 amp.	Hartman Circuit Breaker Co.
2	Switches	Oil circuit-breakers			Electrically operated, 2,200 volts, 3,000 amp.	Hartman Circuit Breaker Co.
2	Switches	Oil circuit-breakers			Electrically operated, 2,200 volts, 600 amp.	Hartman Circuit Breaker Co.
2	Switches	Oil circuit-breakers			Electrically operated, 2,200 volts, 200 amp.	Hartman Circuit Breaker Co.
2	Switches	Oil circuit-breakers			Electrically operated, 2,200 volts, 100 amp.	Hartman Circuit Breaker Co.
2	Starters	Auto		For 425-hp. motors	2,200 volts, 100 amp.	Westinghouse Elec. & Mfg. Co.
2	Starters	Auto		For 720-hp. motors	2,200 volts, 300 amp.	Westinghouse Elec. & Mfg. Co.
2	Transformers		5-kv.-a.	Auxiliary power	2,200-1,100 to 220-100 volts, 54-140 cycles	Westinghouse Elec. & Mfg. Co.

transformers and are tapped for 1,340 volts on the low-tension side with 5 and 10 per cent. taps on both the high- and low-tension windings.

With a star-connected secondary, 2,310 volts applied gives 4,000 volts on the low-tension winding and 23,100 volts on the high-tension side. These transformers are operated with the secondary grounded, so that service may be interchanged with the Wheeling Electric Co., which operates a 4,000-volt, star-connected, grounded neutral system in Wheeling. These transformers act as an auto-transformer between the Wheeling Traction Co. and the Wheeling Electric Co.'s lines.

This work was carried on under the direction of W. E. Moore, general manager, and was superintended by J. S. Jenks, assistant general manager of the West Penn Traction Co. with G. G. Bell as mechanical engineer, J. E. Thomas as electrical engineer, and J. M. Hopwood as superintendent of construction.

Shop and Acceptance Tests of 30,000,000-Gal. Pump

SYNOPSIS—Two tests of a turbine-driven centrifugal pump the results of which check within 2.5 per cent.

During the summer of 1913 Cleveland's demand for water was at times in excess of the capacity of the Kirtland station pumps, and while the reservoirs had sufficient reserve capacity to take care of the peak demands, C. F. Schulz, Commissioner of Water, decided to install temporarily a 25,000,000-gal. pumping unit at the Kirtland station. As the pump was to be moved after two years of operation, a turbine-driven centrifugal unit was the only type considered, as the size, weight and cost of dismantling and reërecting a reciprocating engine would have been excessive when compared with the centrifugal type of unit.

Two pumps connected in series, driven through reduction gears, were selected on account of the increase in efficiency of this arrangement over the single gear-driven pump or the direct-connected single- or multi-rotor pump for a head of 250 ft. The unit will pump against this head when installed later at the Division Avenue station.

Although bids were received on 25,000,000-gal. units, it was decided to equip the pumps with 30,000,000-gal. impellers, as the turbine, reduction gear and pumps were of ample size and the De Laval company, which was awarded the contract, offered to do this without additional expense to the water department.

The principal dimensions, weights and other data of interest of the turbine, pump, reduction gear and condenser are given in Table 1.

The turbine is built with 13 pressure stages, the first pressure stage having two velocity stages. The shaft of the rotor is of such large diameter that it does not pass through the critical speed within the working range of the turbine.

The reduction gear operates very quietly, not only because of the accuracy with which the gear and pinion are cut and lined up, but also because the full-load tooth pressure of 300 lb. per in. of face and the pitch-line speed of 6,070 ft. per min. are quite moderate for gears of this type.

The pump impellers are made of government bronze. They are equipped with wearing rings of the labyrinth type, intermeshing with corresponding stationary rings in the casing. Rings of this type reduce the leakage between the impeller and pump casing to a small amount without the use of very small clearances.

The condenser is of the water-works type and has built into it a condensate or primary heater which has a heating surface of 140 sq.ft. This condensate heater is of special value in a water-works type condenser, as installed at Kirtland station, as the temperature of the cooling water for six months of the year averages less than 40 deg. F.; and since the unit runs at approximately full capacity when in operation, the 30,000,000 gal. of water passing through the condenser tends to cool the condensate to a much lower temperature than is desired and the condensate heater helps materially to bring the temperature of the condensate back to within a few degrees of the vacuum temperature.

TABLE 1. DIMENSIONS AND GENERAL DATA

Turbine Data	
Brake horsepower of the unit at rated pump capacity and 206 ft. total head	1,350
Revolutions per minute at full load	3,510
Number of stages on turbine	13
Net weight of turbine without bedplate, lb.	28,000
Floor space required by the entire unit, ft.	35x7
Reduction Gear	
Type of gear	Double helical
Width of faces of gears, in.	24
Pitch diameter of pinion, in.	6.6
Pitch diameter of gear, in.	39.4
Gear ratio	1 to 6
Spiral angle of teeth, deg.	45
Net weight of reduction gear without bedplate, lb.	23,000
Pump Data	
Rated daily capacity of pumps against 206 ft. total head, gal.	30,000,000
Number of pumps connected in series	2
Revolutions per minute at full capacity	585
Diameter of suction and discharge nozzles, in.	24
Net weight of pump without bedplate, lb.	26,000
Condenser and Condensate Heater	
Make of condenser	W. C. & E. Co.
Diameter of case, in.	62.5
Inside length of case	10 ft. 9 in.
Diameter of tubes, in.	1
Length of tubes	10 ft. 11 in.
Number of tubes	634
Square feet of cooling surface	2,120
Net weight of condenser complete	21,000
Bedplates	
Net weight of turbine bedplate, lb.	8,000
Net weight of main reduction-gear bedplate, lb.	3,800
Net weight of pump bedplate, lb.	3,300
Total weight of unit (not including condensers, etc.), lb.	91,100

The air pump is of the reciprocating dry-vacuum type. A float in the condenser controls the speed of the duplex boiler-feed type condensate pump.

Fig. 1 shows the unit erected in the shop ready for the full-capacity duty test.

The pump at the left takes its suction from a 60,000-gal. concrete tank over which it is erected. It discharges

this water through the series connection into the pump at the right, and this pump discharges the water through the venturi meter and a gate valve used to control the head against which the pumps work, back into the tank. Owing to the design of the tank and to the fact that a venturi meter was used instead of a nozzle, no trouble was experienced with aëration of the water.

TABLE 2. SHOP TEST OF 30,000,000-GAL. DE LAVAL PUMPING UNIT, JULY 7, 1914

General Data	
Length of test, min.	30
Steam pressure above throttle, lb. gage	143.4
Steam pressure below throttle, lb. gage	134.5
Vacuum at turbine exhaust nozzle, hg. in.	26.8
Atmospheric pressure by barometer, hg. in.	29.9
Vacuum at turbine exhaust nozzle corrected for 30-in. barometer, hg. in.	26.9
Suction lift at center line of pump, ft.	15.6
Discharge pressure at center line of pump, ft.	176.2
Total head on pump, including suction lift, ft.	191.8
Rate of pumping as measured by venturi meter manometer, gal. per min.	20,100
Rate of pumping as measured by venturi meter manometer, gal. per day	28,940,000
Pounds of water pumped per minute	167,634
Calorimeter Data	
Temperature of steam in calorimeter, deg. F.	268
Pressure in calorimeter, hg. in.	8.2
Pressure in calorimeter, lb. abs.	18.75
Moisture in steam, per cent.	2.07
Condensate	
Total condensate during test, lb.	8,308
Total condensate per hour, including auxiliaries, lb.	16,616
Duty	
Average work done per minute during test, ft.-lb.	32,157,000
Average water horsepower	974.4
Duty per 1,000 lb. of wet steam, ft.-lb.	116,100,000
Corrections	
Correction for 150 — 143.4 = 6.6 lb. difference between actual and specified steam pressure, assuming 1 per cent. for each 10 lb., per cent.	0.66
Correction for 2.07 per cent. moisture, assuming a correction of 2 per cent. for each 1 per cent. of moisture, per cent.	4.14
Correction for 28.5 — 26.9 = 1.6 in. of mercury difference in vacuum between actual and specified, assuming 8 per cent. for each inch of mercury, per cent.	12.80
Correction for 100 deg. superheat, allowing 1 per cent. for each 10 deg., per cent.	10.00
Total correction, per cent.	27.60
Corrected Duty	
Duty per 1,000 lb. of superheated steam, corrected for vacuum, superheat and pressure, ft.-lb.	148,140,000

When a large volume of water is circulated through a tank holding only 60,000 gal., the use of a nozzle for measuring the quantity pumped is not practical for runs of any considerable length, on account of the rapid aëration of the water, but the use of a venturi meter allows the discharge pipe to be submerged a considerable distance below the surface of the water, which entirely prevents trouble from this source.

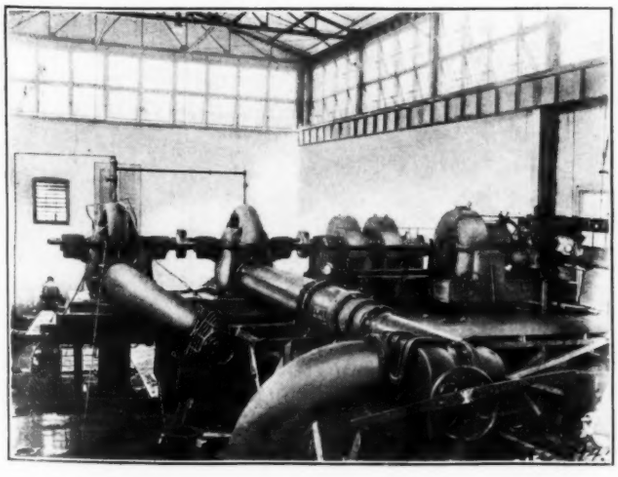


FIG. 1. PUMP AS ARRANGED FOR SHOP TEST

All instruments used in making the test were carefully calibrated. The principal results of the shop test are given in Table 2.

The fact that the duty obtained on this test checked within 2½ per cent. of that obtained on the official test after installation, in spite of the very considerable difference in quantity of water pumped, total head and steam conditions, indicates that the corrections used for vacuum, moisture and superheat are very nearly correct for this type of turbine. The amount of these corrections is given at the end of Table 2. As the turbine and pump efficiency curves are quite flat for a considerable range each side of full load, about the same duty should be expected on both tests in spite of the fact that the unit worked under a load of 974.4 water hp. on the shop test and 1,120 water hp. on the official duty test.

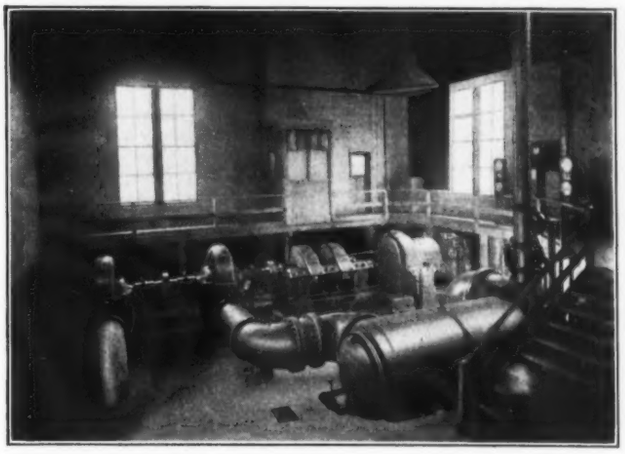


FIG. 2. PUMP AS ARRANGED FOR ACCEPTANCE TEST

A shop test like the foregoing is of value not only because it assures the purchaser that the unit will meet the manufacturer's guarantees before it is shipped, thus doing away with the undesirable complications that arise after installation when the manufacturer cannot meet his capacity or efficiency guarantees without changing pump impellers, turbine nozzles or such like, and thus forcing at least part of this expense and loss of time on the purchaser, but it also gives the manufacturer an opportunity to test out the design and construction of the apparatus under the direction of his engineering and shop experts and allows the remedy of practically all defects before the unit is shipped to the customer, whereas in many cases where shop tests are not made, the unit is tested and erected by an erecting engineer or roadman who is not interested in the details of design or construction and whose main idea is to get the unit going and turned over to the purchaser as soon as possible.

Fig. 2 shows the unit installed in the temporary engine house where it has now been in operation over a year. The water-works type of condenser was not installed below the turbine, as this would have been an unnecessary expense for this temporary installation.

After installation the unit was operated for several weeks to give it a thorough try-out before the official acceptance test was run. The results of the duty trial are shown in Table 3.

The water discharged by the pump was measured by two venturi meters, a 36-in. and a 48-in. The registers

of these meters can be seen by the window at the right in Fig. 2. On the duty trial the readings of these meters checked within $\frac{1}{10}$ per cent. of each other, and since there is quite a difference in their size, the fact that they checked so closely with each other and also with a meter of still a different size on the shop test would indicate that they

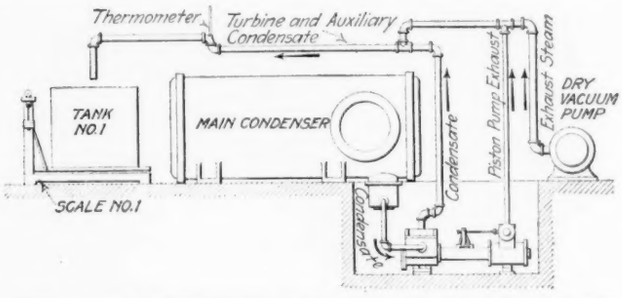


FIG. 3. ARRANGEMENT FOR CONDENSING AUXILIARY EXHAUST AND WEIGHING CONDENSATE

measured with very little error the water passing through them.

The condensate was weighed in two tanks on calibrated scales after it had been heated by the exhaust steam from the turbine and the dry-vacuum and condensate pumps.

The method of condensing the auxiliary exhaust steam is shown in Fig. 3.

The vacuum in the condenser and the suction lift of the pump were measured with mercury columns. All other readings of pressure were taken from gages calibrated with a dead-weight gage tester.

A comparison of the principal results of the test and the guarantees made are given in Table 4.

The guarantees were made on the B.t.u. basis. All steam supplied to the turbine and its auxiliaries was charged against the unit, but the unit was credited with all the heat above 32 deg. F., that was returned to the weighing tanks by the auxiliary exhaust steam and turbine condensate, as this was available for boiler-feed water.

No bonus was given the contractor for exceeding the guaranteed duty, which was as high as that of any of the other bidders and was capitalized on the basis of $500 for each million foot-pounds and the capitalized value applied to the prices bid on the unit in awarding the contract.

TABLE 3. RESULT OF DUTY TRIAL ON 30,000,000-GAL. DE LAVAL PUMPING UNIT NO. 1, KIRTLAND PUMPING STATION, CLEVELAND, OHIO

Date of trial	Dec. 22, 1914
Duration of trial, hr.	5
Average Pressures	
Atmospheric pressure by barometer, lb.	14.37
Steam pressure at throttle, lb. gage	149.9
Steam pressure at throttle, lb. abs.	164.3
Steam pressure at nozzles, lb. gage	144.6
Vacuum in condenser (corrected), in.	28.47
Water pressure at discharge nozzle of pump, ft.	188.1
Suction lift at center line of pump, ft.	21.8
Total average head on pump, including suction lift and friction, ft.	209.9
Average Temperatures	
Average temperature of water leaving condenser, deg. F.	38.0
Temperature rise through condenser, deg. F.	3.0
Average temperature of condensate leaving condensate heater, deg. F.	82.0
Average temperature of condensate at weighing tanks after being heated by auxiliary exhaust, deg. F.	102.2
Average temperature of steam at throttle, deg. F.	440.3
Quality and Quantity of Steam	
Superheat in steam supplied to turbine and auxiliaries, deg. F.	74.6
Total steam consumption per hour of turbine and auxiliaries, lb.	14,998
Equivalent total heat consumed by turbine and auxiliaries per hour during trial, thousand B.t.u.	17,499
Pumpage and Work Done	
Average revolutions per minute of pump	580.2
Average rate of pumping during trial by venturi meter manometer, million gal. per day	30.38
Total work done by pumping unit per hour, million ft.-lb.	2,218
Average water hp.	1,120.4
Duty	
Duty per million B.t.u., ft.-lb.	126,780,000
Duty per 1,000 lb. superheated steam, ft.-lb.	148,120,000
Steam per actual water hp. per hr., lb.	13.37
Thermal efficiency, per cent.	16.3
Duty per 1,000 lb. steam corrected for 100 — 74.6 = 25.4 deg. of superheat to compare with shop test, ft.-lb.	151,900,000

TABLE 4. TEST AND GUARANTEED DATA

	Specified and Guaranteed Data	Results Obtained on Duty Trial
Pumpage rate, gal. per day of 24 hr.	30,000,000	30,380,000
Suction lift to center line of pump, ft.	18	21.8
Total head, ft.	206	209.9
Steam pressure, lb. gage	150	149.9
Superheat, deg. F.	100	74.6
Duty per 1,000,000 B.t.u., ft.-lb.	115,500,000	126,780,000
Duty per 1,000 lb. of steam, ft.-lb.	136,500,000	148,120,000

JUST FOR FUN

Two More "Fool Stunts"

An engineer (?) who was told to mix putty and red lead and stick a little of it on the piston and roll the engine over to determine the clearance, took the cylinder head off and carefully painted the piston red, put the head back on and ran the engine all night, but next morning he did not know any more than before what the clearance was. It is not clear what he expected to find.

An engine with a Sweet valve leaked some, and the engineer wanted to let the pressure plate down two or three thousandths of an inch. To do this he used a cold chisel and hammer to cut away the pressure plate where it rested on the shoulders in the steam chest. Words fail me!—Harry D. Everett, Washington, D. C.

Label It "This Is a Dynamo"

While a shift engineer in a 3,000-hp. traction plant, the chief introduced me to Professor ——, who was then lecturing on electricity at one of the town's schools.

I was told to show him round and explain everything very fully; so I took him over to the engines, explaining the different types of valve gears; then to the switchboard, and in turn to the dynamos, mounted on the shaft between the high- and low-pressure engines, and as fully as I knew explained them to him.

PROFOUND THOUGHT

After showing him the boilers, explaining the action of mechanical stokers, economizer, etc., I said: "Now we'll go and look at the car-repair department." Thereupon he tapped me on the shoulder and said: "Wait a bit, young fellow. Show me now, will you, where you generate the electricity." I told him that we had just come from the engine room where he had seen the dynamos. "Were those the dynamos? I didn't realize it. Come back, I'm going to have another look." So it was the whole place over again.—H. Baron, Stettler, Alta.

Heating by Exhaust Steam

SYNOPSIS—The article shows graphically the comparative amounts of heat thrown away when an engine is run condensing and the rejected heat allowed to go off in the overflow from a condenser, or when the engine is run noncondensing and more or less of the heat in the exhaust is applied to useful purposes.

In the accompanying diagram vertical distances are proportional to absolute temperatures and areas are proportional to quantities of heat.

Starting with a pound of water at 32 deg. F. = 492 abs., and raising it to 360 deg. F. = 820 abs., there are put into it 331.9 B.t.u., represented by the area *ADILA*. At 360 deg. (153 lb. absolute) the water boils, and 861.8 B.t.u., represented by the area *DEHID*, are taken up in its evaporation to dry saturated steam, the temperature remaining constant along the line *DE*, which is made of such length that the area under it will represent the heat of evaporation *L*, 861.8 B.t.u., to the same scale that the area under the curve *D* represents the heat of the liquid *h*. This length *DE* is proportional to the entropy of evaporation at that temperature, the length of the line *PD* to the entropy of the liquid of the same temperature, and the length of the line *PE* to the total entropy of dry saturated steam at 360 deg. The values of all these entropies for different pressures and temperatures may be found in the steam tables.

The total heat necessary to make a pound of dry saturated steam at 360 deg., out of a pound of water at 32 deg., is then 331.9 + 861.8 = 1193.7 B.t.u., represented by the area *ADEHLA*.

Suppose the steam is worked in a condensing engine or turbine between the limits 360 and 100 deg. The utmost that could be converted into mechanical energy by an engine of 100 per cent. efficiency would be the heat represented by the area *BDEGB;* the rest, represented by the area *BGHKB*, would go out (radiation neglected) in the exhaust.

The area *ABKLA* is out, because it is assumed that feed water of the exhaust temperature can be had, so that the reheating process starts at 100 deg. *B* instead of 32 deg *A*.

The utmost efficiency of the cycle, even with 100-per cent. apparatus, would be

$$\frac{BDEGB}{BDEHKB} = \frac{1193.7 - 67.97 - 805.34}{1193.7 - 67.97} = 28.46 \text{ per cent.}$$

The heat put in (the denominator) is proportional to the area *BDEHKB*, which is less than that of the whole area down to the 32 line by the area *ABKLA*, proportional to the heat of the liquid at 100 = 67.97.

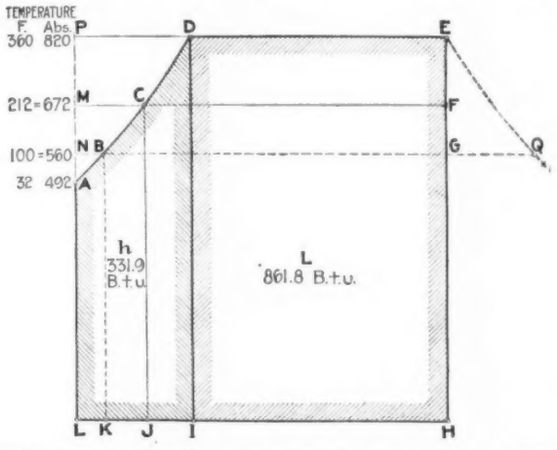
The heat converted into work (the numerator), proportional to the area *BDEGB*, is determinable by subtracting from the heat represented by the whole area *ADEHLA* that represented by the area *ABKLA* and that represented by the rectangle *BGHKB*. The length of the latter is *BG* = *NG* — *NB;* the line *NG* is proportional to the total entropy of steam at 360 deg. F. = 1.5676; *NB* to the entropy of the liquid at 100 deg. = 0.1295: *BG* to their difference, 1.5676 — 0.1295 = 1.4381; the height *GH* is proportional to the absolute temperature at 100 deg. F. = 100 + 460 = 560 deg. The area = length × height = *BG* × *GH* = 1.4381 × 560 = 805.34.

The value in B.t.u. of the convertible area *BDEGB*, therefore, is 1193.7 — 67.97 — 805.31 = 320.39 B.t.u., and the theoretical efficiency of the cycle,

$$\frac{320.39}{1193.7 - 67.97} = 28.46 \text{ per cent.}$$

In this process all the heat represented by the area BGHKB, *71.54 per cent. of that furnished is thrown away, in order that the heat represented by the area* BDEGB *may be converted into mechanical work.*

If, instead of running the engine condensing with an exhaust at 100 deg. F., it be run with exhaust at atmospheric pressure (212 deg.) only the area *CDEFC* (169.5 B.t.u. = 16.72 per cent.) of the heat furnished can be converted into mechanical energy, but the rest of the heat (represented by the area *CFHJC*) is at a sufficiently high temperature to be useful for heating and

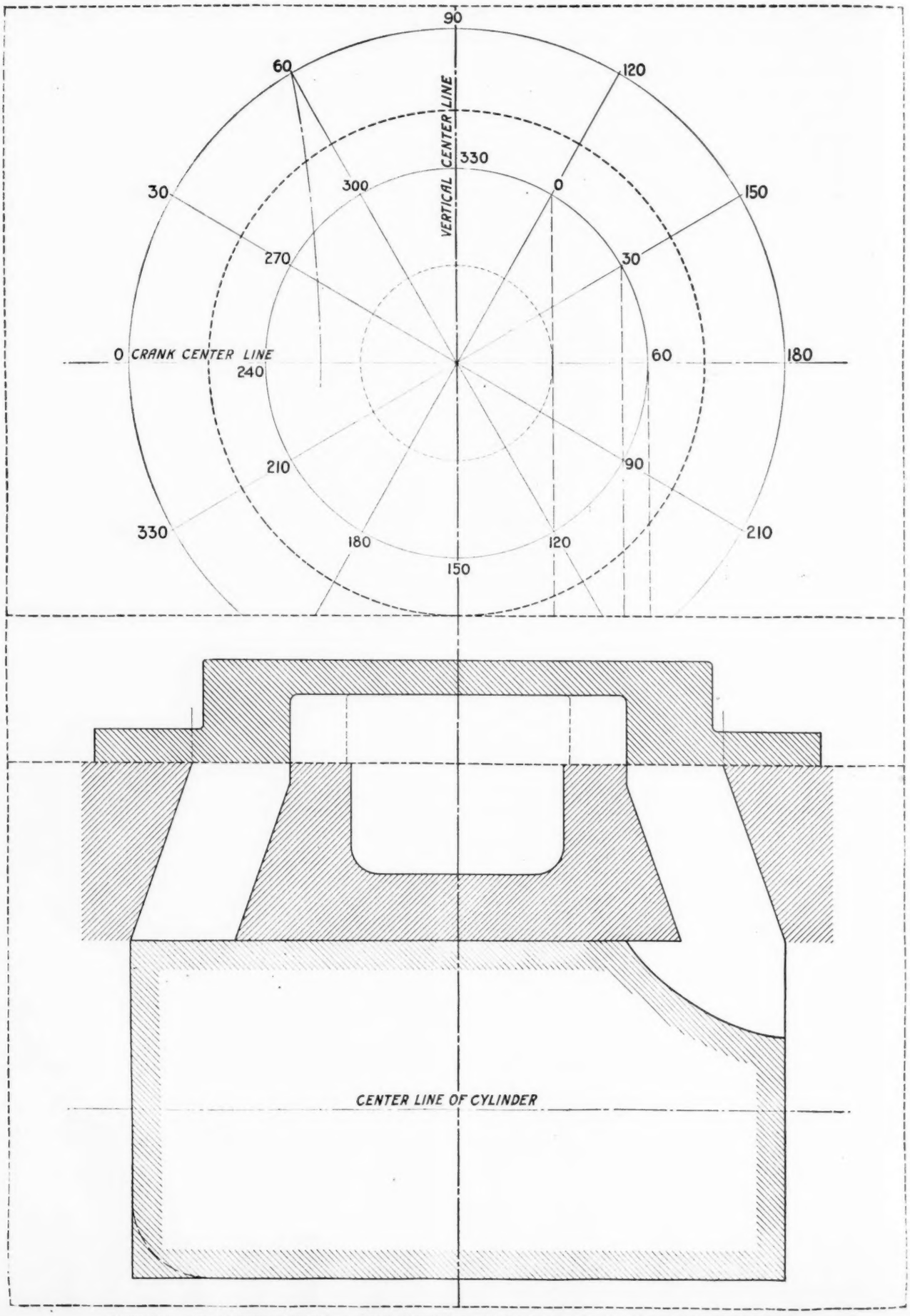


SHOWING HEAT CONVERTIBLE INTO WORK AND THAT AVAILABLE FOR HEATING

industrial purposes, and if it can all be condensed back to water at 212 deg., all of the heat used in making the steam will have been devoted to some useful purpose, giving (radiation neglected) a 100-per cent. proposition.

If there is any useful purpose to which the heat represented by the area *CFHJC*, or a proportion of it considerably larger than that represented by the area *BCFGB*, can be applied, it is not good engineering to throw it away in the discharge of a condenser, in order to rescue the comparatively small amount represented by the area *BCFGB*.

Electrolytic Tests made by the engineers of the New York Board of Water Supply to determine the danger of corrosion to the Bryn Mawr siphon of the Catskill Aqueduct indicated that the steel pipe carried several volts from trolley tracks at Tuckahoe Road to the Bryn Mawr connection chamber. Here the current left the steel pipe through a 2-in. line supplying the hydraulic cylinders of the valves, returning through the Yonkers water mains to the railway company's substations. As electrolytic action always takes place where the current leaves the conductor, and as the current passes directly from the siphon to the 2-in. pipe without passing through any earth, it was decided that the steel siphon was safe from electrolytic corrosion.—"Engineering Record."



METHOD OF CONSTRUCTING A THEORETICAL INDICATOR DIAGRAM BY MEASUREMENT

Will Quizz, Jr.

SYNOPSIS—Chief Teller gives Will a tryout on some of the things they have talked over and outlines a means of applying a critical steam distribution test to an engine.

"I wish, Will, you would make a diagram from No. 3 engine for me today."

"Why, Chief, No. 3 is down for repairs, the piston is out, the valve cover is off—"

"That's all right, Will; with the valve-chest cover off you can make the necessary measurements without extra labor."

"But, Chief, I do not understand what you are getting at, since you want that diagram today."

"It is just this, Will. I want you to take advantage of this opportunity to apply the knowledge you have on valve motions and the action of steam, etc. Go and take the valve out, measure it and the ports carefully, use a measuring stick and mark the edge of the ports on it. In order to save yourself some trouble, first find out the extreme travel of the valve. It will be well enough to note approximately the position of the crankpin when the valve motion is at each extreme of its travel. When you have done this, take three pieces of smooth pine board and lay out the ports on the edge of one, the valve on the edge of the second and the crank and eccentric circles on the third. Then establish center lines across each, which will register with one another when the valve is in midposition, and with other marks as the valve moves to its various positions, and in this way work out the kind of a diagram to be expected. Now, I think you can see what I am getting at."

"Yes, Chief, I do. You want me to construct a theoretical diagram for that engine."

Perfect Diagrams Used for Comparisons

"You're hep. After all the pages in the papers and the books that have been written on this subject, there are a great many engineers who do not realize their full meaning. For instance, they will admire a well formed Corliss diagram and wonder why the ones produced by a high-speed shaft governing engine do not look the same. Model diagrams in general are all very well, but the theoretically perfect diagram from a given engine is vastly superior when it comes to judging of that engine's regular performance. It amounts to about the same as taking any prescription the druggist may have on file, or having a reputable physician make a careful examination and prescribe for an ailment on that basis.

"You can get the idea from this illustration, [on opposite page], which may be pasted on cardboard and cut. The outline of a common slide valve is used, but the principles are the same for all valves. The ports are of about the ordinary proportion. The slide valve, if made to just cover both ports (as shown in its central position), or cut off at the lines at the edges of the ports, would be termed a valve without inside or outside lap, because it did not overlap the ports. The circles with their common center on the vertical center line represent the engine-crankpin travel and the eccentric or governor-pin travel, the larger and smaller ones, respectively.

"The smallest, or broken, circle would be all the travel the valve would require for full port opening if it had no lap. In that case the eccentric would be at right angles to the crank. Suppose the crankpin to be at zero and the eccentric at 90 deg., or on the vertical line. Then when the crank advanced to 90 deg. the eccentric would be at its extreme travel, and when the crank reached 180, or the end of its stroke, the eccentric (and valve) would be back to the central position and cutoff would not have occurred until the end of the stroke.

Effect of Adding Outside Lap

"This condition is not often desirable, so we will add to the valve on both ends (called outside lap). Now you see, the port would hardly uncover at all when the eccentric was at its extreme travel, so we must use an eccentric with more throw (see the next larger circle). With the eccentric in the same position—90 deg. ahead of the crank—there would be no steam entering at the first of the stroke. This, of course, won't do, so we must rotate the eccentric ahead far enough to get the desired 'lead,' or port opening, when the crank is still at zero. This is the angle of advance of the eccentric. In this case it is 30 deg. (120 deg. ahead of the crank) when the port begins to open. Shift the valve so that the center line on the valve registers with a line dropped from the circle at the 120-deg. radius and call this point zero for the eccentric. Now follow the motion of both crank and eccentric, as they rotate in unison, and move the valve forward and back to register with the vertical lines from these points. The progress of the eccentric is from zero toward 60 deg., which marks its extreme while the crank is making similar progress; then on its return stroke to 120 deg., at which point the valve is closed. The crank at the same time has reached the 120-deg. point in its travel when the steam is cut off. Drop a vertical line from 120 deg. on the outer circle to the space representing the cylinder, and the point of cutoff will be established (approximately) for the diagram.

"Now, for an earlier cutoff the lap must be increased and the eccentric advanced as before, and in practice the engine is given considerable port opening or 'lead' at the beginning of the stroke, and the valve travel is somewhat restricted and does not give full port opening at light load and early cutoff, because for such conditions full port opening is not necessary.

Extremes in Lap and Eccentric Changes

"It might easily be possible, with a valve of the proportions shown, to reduce the valve travel to nearly that of the inner circle and then increase the angle of advance to 180 deg. of the crank circle, so that the amount of lead would be the total port opening and the valve would begin to close at the beginning of the crank's stroke. This is what actually occurs with some forms of valves and automatic governors when running up to speed with no load. This feature, I hope we will be able to take up some other day. After adding outside lap and advancing the eccentric, you will see that the exhaust port closes as many degrees earlier as the eccentric is advanced. If you wanted still earlier closure (more compression) lap would have to be added on the inside of the valve (called inside lap). With a valve of the proportions shown and the

eccentric advanced 30 deg., the exhaust port would be closed when the crank was at the 150- and 330-deg. points, giving only a slight compression, somewhat as shown in the lower left-hand corner. If, however, the angle of advance is increased, as before mentioned, the exhaust port would close and compression begin at about half stroke, so you see changing one event changes all the others after the dimensions are once fixed.

"By constructing a model from any engine, with the ports and valve measurements true, you can learn more of its action and the kind of diagram it should produce than in any other way that I know of."

"But, Chief, this gives me only the valve motion. How should I proceed to get the diagram?"

"Drop vertical lines from the various positions of the piston relative to the crank and corresponding to the same number of degrees traveled by the eccentric to complete any of the various events.

Establishing the Piston's Position

"To get the exact position of the piston for each point or degree of the crank circle (which will be necessary when you come to construct diagrams with different points of cutoff, release, exhaust closure, etc.) without the necessity of calculating the angle of the connecting-rod, extend the horizontal center line through the circle far enough to equal the relative length of the connecting-rod (five to six times the length of the crank of the engine is about the ordinary length of the connecting-rod, but measure the one you are working on to get it right). If the connecting-rod on the engine is six times the length of the crank, then three times the diameter of the crank circle will be the proper length for your model. With a measuring stick of this length or, better still, a large compass, place one point on the crank circle at the desired degree (say 60) and the other out on the horizontal center line wherever it may reach to, then bring the first point down to the center line within the circle (without moving the other point). This will give the relative position of the piston, or the horizontal distance it has traveled. Vertical lines dropped from points so established will give you the various points desired for the diagram. This we had several months ago (June 16, 1914). The space below the valve representing the cylinder, in which the diagram is to be drawn, helps by showing graphically just the pressure acting on the piston at any point of the stroke.

"When the point of cutoff has been established and no more steam is admitted, then the expansion curve is plotted according to the law of the expansion of a perfect gas, known as Boyle's or Mariotte's law (described Dec. 15, 1914). For instance (disregarding the clearance), if cutoff occurred at one-quarter stroke, then at one-half stroke the pressure would be half that of the initial, and at the end of the stroke it would be one-quarter of the initial, and at intermediate points in proportion to the space the steam occupied.

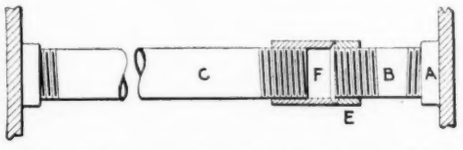
"This I want you to work out for yourself. Make a model like the illustration, and let me know how you make out with it. If I am not mistaken you will find it as interesting as I did at one time, and still find it so for that matter. Do not consider it a mere toy, however. In fact, engines are designed and built around a diagram, and not simply a diagram made to see how it looks after the engine is built."

Some Dangerous Long Screws

By Lewis F. Lyne

I have frequent use for what is known as a "Long Screw" in replacing 2-in. extra heavy wrought-iron pipes in water arches that are either worn out or damaged by incrustation or scale. These pipes range in length from 20 to 48 in. and are held at each end by fixed cast-iron manifolds. After the damaged pipe has been cut out with a hacksaw and the ends removed from the threaded holes, a long screw nipple is inserted in one hole *A* and a long pipe cut to fill in the gap between this casting and the opposite side and the long screw. To insure a good job and tight joints, the pipe ought to screw into the casting at least 1 in., which would leave a 1-in. gap at *F*. The couplings used on standard long screws are about 2 in. long, therefore there would be no threads left on the long screw inside the coupling, and it is impossible to make a joint by bringing up the follower *E*. There are thousands of such long screws in use, many of which are unsafe. A standard (not patented) long screw for this kind of work has come into use extensively. This has an extra long coupling and there has not yet come to me a single complaint of its failure.

There is another little "trick" in connection with the use of long screws—the entering of the threads on the pipe *C* into the coupling. If both pipes are screwed up tight and the coupling forced upon the threads on the



EXTRA-LONG SLEEVE ON LONG SCREW

long pipe, they may fit or they may not; probably not. They may push, or if they do not match within a half thread, they may be stripped altogether, and this often happens. The threads ought not to be forced together. The most reliable method of matching the threads in the coupling to those on the pipe is as follows: Slack up the nipple *B* (if it is a right-hand thread) about one turn, and then try to catch the coupling onto the long pipe. If it does not readily enter, slack the nipple another fraction until the threads match, after which the nipple may be screwed in again to its original position. These directions apply only when the thread on the nipple in the casting is a right-hand; if it is not, then the adjusting must be done with the pipe *C*. Always adjust with a right-hand thread.

In ordering wrought-iron pipes, unless the order plainly specifies and emphasizes the term W. I., the average dealer will ship steel pipe. The leaks with steel pipe in this particular service are more than double those of wrought iron, due principally to poor threads cut with dull dies. Steel dulls the dies more quickly than iron pipe and therefore tears the threads; hence the bad fit and leak. Dies used alternately to cut iron and steel pipe do not work well and may be compared to the use of files, on brass and iron, for every machinist knows that a new file is necessary to cut brass, and after it has been used on iron, it will not take hold of brass satisfactorily. A long screw with an extra long coupling, as described, when used on clean cut threads makes a good joint.

Using a Slide Rule

SYNOPSIS—Simple directions for manipulating the rule to work out problems in multiplication, division and proportion. Practice examples are given, and assuming the reader with rule in hand, he is told how to set the values and where and how to read the answers.

One of the simplest uses of the slide rule is to solve proportions. You know what a proportion is:

$$2 : 6 :: 3 : 9;$$

as 2 is to 6 so is 3 to 9; or $\frac{2}{6} = \frac{3}{9}$.

As you know, there are four scales on the rule, the A-scale at the top, below it the B-scale, the D-scale at the bottom and above it the C-scale. Set your rule with the 2 of the C-scale over the 6 of the D-scale, and you will see that all the values on the C-scale bear the same ratio to those on the D-scale that 2 does to 6. Not only is the 3 over the 9, but everything on the scale is in that proportion. Read them along

$\frac{1}{3}$ $\frac{11}{33}$ $\frac{12}{36}$ $\frac{13}{39}$ $\frac{14}{42}$ $\frac{15}{45}$ $\frac{16}{48}$ $\frac{17}{51}$ $\frac{18}{54}$ $\frac{19}{57}$ $\frac{20}{60}$ $\frac{21}{63}$ $\frac{22}{66}$, etc.

Put the right-hand index or 1 of the C-scale over the 3 of the D-scale, where the left-hand index now is, and you can read the rest of the scale

$\frac{4}{12}$ $\frac{45}{135}$ $\frac{5}{15}$ $\frac{55}{165}$ $\frac{6}{18}$, etc.

All that you need to do then to solve a proportion is to place the first term of the proportion on the C-scale over the second on the D-scale, and under the third term on the C-scale will be found the fourth term on the D-scale.

$$\begin{matrix} C & D & C & D \\ 2 & : & 6 :: 3 & : & 9 \end{matrix}$$

Example: An engine develops 76.5 hp. running at 130 r.p.m. What horsepower would it develop with the same mean effective pressure running at 140 r.p.m.?

The power is directly proportional to the speed, hence

$$\begin{matrix} C & D & C & D \\ 130 & : & 140 :: 76.5 & : & x \end{matrix}$$

Set 130 on the C-scale over 140 on the D-scale and under 76.5 on the C-scale find the answer 82.4.

Example: A pulley 36 in. diameter running 275 r.p.m. is belted to one 20 in. diameter. How fast does the smaller pulley run?

Here the speeds will be inversely as the diameters. The smaller its diameter the faster the pulley will run. When you state a proportion, put the given quantity which is of the same kind as the answer, as the third term. In this case the answer is in revolutions per minute. The given term of this kind is the 275 r.p.m. of the larger pulley. Write this as the third term. Then consider whether the answer should be larger or smaller than this third term. If it should be larger, place the larger of the other two given terms as the second and the smaller as the first term. If it should be smaller, place the smaller of the other two given terms as the second and the larger as the first. In this way you will be right whether the proportion is direct or inverse. In this case the number of revolutions of the smaller pulley will obviously be greater than those of the larger pulley, which drives it, so we have

$$\begin{matrix} C & D & C & D \\ 20 & : & 36 :: 275 & : & 495 \end{matrix}$$

Many of the ordinary calculations may be stated as proportions and thus easily solved. The same setting of the rule shows that if we want the small pulley to run 500 r.p.m. the larger must run about 278, etc.

The product of the means, or two inside terms of a proportion, is equal to the product of the extremes or two outside terms:

$$\begin{matrix} 2 & : & 6 :: 3 & : & 9 \\ \text{(means)} & & & & \text{(extremes)} \\ 6 & \times & 3 & = & 2 \times 9 \end{matrix}$$

To find either mean it is necessary only to multiply the extremes together and divide by the other mean; or to find one extreme, to multiply the two means together and divide by the other extreme. Thus if we have given

$$\begin{matrix} C & D & C & D \\ 2 & : & 6 :: 3 & : & x \end{matrix}$$

we can find the unknown term x thus:

$$\frac{\begin{matrix} D & C \\ 6 & \times & 3 \end{matrix}}{\begin{matrix} 2 \\ C \end{matrix}} = \begin{matrix} D \\ 9 \end{matrix}$$

Set the divisor (2) on the C-scale over the first multiplier (6) on the D-scale, and under the second multiplier (3) on the C-scale read the answer (9) on the D-scale. The same process is therefore good for ordinary multiplication and division.

Example: A boiler evaporates 8.5 lb. of water per pound of coal. The factor of evaporation is 1.15; that is, the evaporation of one pound under the conditions of the test is equivalent to the evaporation of 1.15 lb. from and at 212 deg. F. The coal has 8 per cent. of ash. What is the evaporation from and at 212 deg. F. per lb. of combustible?

If 8 per cent., or 0.08, of it is ash, one pound of the coal contains 1 — 0.08 = 0.92 lb. of combustible. The problem then stands

$$\frac{\begin{matrix} D & C \\ 8.5 & \times & 1.15 \end{matrix}}{\begin{matrix} 0.92 \\ C \end{matrix}} = \begin{matrix} D \\ 10.62\ lb. \end{matrix}$$

Set 92 on C over 85 on D, and under 115 on C read the answer, 10.62 on D.

Sometimes you may have to shift the slide to get the final reading. Suppose you had

$$\frac{\begin{matrix} D & C \\ 32 & \times & 21 \end{matrix}}{\begin{matrix} 70 \\ C \end{matrix}} = \begin{matrix} D \\ 9.6 \end{matrix}$$

Set the 70 on C over the 32 on D, then below 21 on C should be the answer. But there is nothing below 21 on C. Put the hair line of the runner over the right-hand 1 of the C-scale. With the runner in this position, slide the C-scale to the left until the left-hand 1 is under the hair line. Then under the 21 of the C-scale read the answer 9.6. You can place the decimal by using a little common sense. It is evident that the answer will be $\frac{3}{7}\frac{2}{0}$ (a little less than one-half) of 21. This is obviously 9.6 and not 96 or 0.96. It is not always easy to locate the point by inspection. Some helpful rules for locating the decimal point were given in *Power*, Apr. 11, 1911, page 551.

A rule made with the C- and D-scales repeated on the top half in place of the shorter scales of the Mannheim

rule, but with the index in the center avoids the necessity of shifting the slide. You can always get your reading on either the top or bottom scale, whatever the setting may be.

For the simple multiplication of two numbers, you can use the same form, putting 1 in the denominator, thus

$$\frac{\underset{\text{C}}{1} \times \overset{\text{C}}{72}}{\underset{\text{C}}{1}} = \underset{\text{D}}{4{,}896}$$

Set 1 on C over 68 on D and under 72 on C read the answer on D. It will be seen to be almost 490, and as $2 \times 8 = 16$, the last figure must be 6, so it is obviously 4,896. In choosing the 1 on C, use that which will leave the D-scale under the other multiplier. You are taking $\frac{68}{1}$ths of 72 instead of $\frac{68}{144}$ths or some other fraction.

For the simple division of two numbers use the same form again with a 1 in the numerator instead of the first multiplier, thus

$$\frac{\underset{\text{C}}{1} \times \overset{\text{C}}{72}}{\underset{\text{C}}{12}} = \underset{\text{D}}{6}$$

Set 12 of the C-scale over 1 of the D-scale and under 72 of the C-scale find the answer 6 on D. You have simply taken $\frac{1}{12}$ of 72. Use the index of the D-scale which will leave some of the C-scale above the dividend.

Practice proportion, multiplication and division in this way. Use your rule often, checking its readings by computation until you get facility with and confidence in it.

Relative Costs of Coal and of Oil Fuels

By F. C. Fearing*

SYNOPSIS—The chart shows the relative cost of dry fuels and oil fuels in terms of either the cost per boiler-horsepower-hours, or of the evaporation from and at 212 deg. F. produced at a cost of 1c. It is useful in connection with the splendid charts for coal alone, as recently published in "Power." The efficiency of boiler and grate is taken strictly to mean the average overall fuel efficiency and has nothing to do with the evaporation that might be obtained when the plant is "spruced up."

In plotting the curves for oil it was found expedient to assume a "typical" fuel oil. The average B.t.u. per gallon on four oils that vary greatly in character and gravity was found to be 142,214, and a gallon of this oil was found to weigh 7.4849 lb., corresponding to approximately 26 deg. Bé. gravity.

The coal and dry-fuel curves, although plotted for convenience on the same set of axes, must be looked upon as independent of the oil curves.

The latter was plotted from the following formula:

$$\textit{Fuel cost in dollars per b.hp.-hr.} = \frac{(33{,}479\ \textit{B.t.u.}) \times (\textit{cost of oil per gal. in dollars})}{(\textit{B.t.u. of oil per gal.}) \times (\textit{overall efficiency of boiler and furnace})} \quad (1)$$

in which 33,479 B.t.u. is the amount of heat required to be usefully absorbed to produce 1 b.hp. and in which 1 b.hp.-hr. is taken to mean an evaporation from and at 212 deg. F. of 34½ lb. of water per hour.

The formula for coal is approximately the same:

$$\textit{Fuel cost in dollars per b.hp.-hr.} = \frac{(33{,}479\ \textit{B.t.u.}) \times (\textit{cost per gross ton burned})}{(\textit{B.t.u. per lb. of coal available}) \times (2{,}240\ \textit{lb.})} \quad (2)$$

The values of the cost per b.hp.-hour were plotted against different prices per gallon, and for overall efficiencies ranging from 55 to 100 per cent. The cost of 1 b.hp.-hr. will be found along the under side of X axis; on the upper side may be found the number of pounds of water, which, at a cost of 1c., may be evaporated from and at 212 deg. F.

The evaporation from and at 212 deg. F. per lb. of oil fired may be obtained by projecting the intersection of the oil-efficiency curve with the broken curved line called "B.t.u. of oil fuel absorbed at various efficiencies," onto the Y axis.

Example for Oil at 85-Per Cent. Efficiency

To find the cost per boiler horsepower-hour of oil fuel at 85 per cent. efficiency, the oil costing $0.023 per gal. burned, proceed as follows:

From the $0.023 point on the scale of costs per gallon (on the axis of Y'), project parallel to the axis of X until the projection intersects the 85 per cent. oil-efficiency line. Call this point P. A perpendicular from this point to the base-line at once gives the result on either of two ways. In this case we find that the cost per boiler horsepower-hour, is $0.00635 and is equivalent to an evaporation of approximately 54.2 lb. of water at a cost of 1c.

In addition to the foreging, by projecting to the left from the point P, find that the evaporation per pound of oil fired is approximately 16.48 lb. "from and at," and that 16,000 B.t.u. per lb. of fuel must be absorbed to accomplish this.

Let us test these answers:

$$\textit{B.hp. per gal. oil fired} = \frac{(142{,}214\ \textit{B.t.u. per gal.}) \times (0.85\ \textit{efficiency})}{33{,}479\ \textit{B.t.u.}} = 3.61$$

A gallon of oil, however, costs $0.023, which, divided by 3.61 equals $0.00637, the cost of 1 b.hp.-hr., which checks closely the result found by use of the chart.

As the heat value of coal varies between wide limits and is not a comparatively stable quantity, as with oils, the coal curves had to be plotted so as to represent various prices per long ton as burned, instead of the various efficiencies taken for the oil curves. The coal efficiencies are then obtained from the set of supplementary curves, as explained hereafter. To facilitate operations it is assumed that the coal contains only those heat units usefully absorbed at the given efficiency; thus a coal con-

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taining 15,000 B.t.u. and burned at 60 per cent. efficiency is assumed to be equivalent to a coal containing only 9,000 B.t.u. and burned at 100 per cent. efficiency.

The coal curves are plotted against B.t.u. per pound on the scales for coal, which will be found on the right of the chart along the inside of the Y' axis.

Let us work an example:

Assume that we are burning a coal containing 13,000 B.t.u. at an overall efficiency of 60 per cent., the coal costing $3.50 per long ton consumed. Starting from the 13,000-B.t.u. mark on to Y' axis, follow the light dotted slanting line until it intersects the 60 per cent. perpendicular from the base line. From the point thus found, project right or left, as may be necessary, to the $3.50 coal curve. Call this point P'.

The perpendicular to the base line from P' indicates that the cost of 1 b.hp.-hr. is approximately $0.0067, in which the last figure is estimated. This we also find is equivalent to an evaporation of approximately 51 lb. of water "from and at," at a cost of 1c.

By carrying the projection from the point P' over to the Y axis, we find that the evaporation "from and at" is 7.23 lb. and that approximately 7,790 B.t.u. must be absorbed.

Check the accuracy of this result:

$$\frac{(2{,}240\ lb.) \times (13{,}000\ B.t.u.) \times (0.6)}{(33{,}479\ B.t.u.)} = 521\ B.hp.$$

produced from one ton of coal. Therefore, dividing the cost per ton by 521, find that the cost of 1 b.hp.-hr. equals $0.00671 and that the chart result represented an error of 7 in the third place of digits, or only a trifle over 1 per cent.

From the two cases of oil and coal set forth, we see that for the assumed condition, the fuel cost per 100 b.hp.-hr. for oil fuel will be 63½c., as against 67.1c. for coal.

Oil fuels other than the one plotted require, after the evaporation "from and at" has been found directly from the chart, that a correction be applied to the oil-efficiency curve first used, before the boiler-horsepower-hour costs can be estimated, for the following reasons:

Oil is sold by the gallon. Some heavy oils yield more heat units per gallon than the lighter ones, even when the heat values per pound may be somewhat less. Some recent investigations have tended to show that even the heavy oils may be considered as having 19,000 B.t.u. or more per pound. Consequently, oils heavier than 26 deg. Bé. (the oil plotted) will weigh more per gallon, and therefore a gallon of such oil will be more efficient than a gallon of the lighter kind. It should be pointed out that although a greater evaporation per gallon may be obtained from heavier oils, yet no greater evaporation per pound will necessarily be effected.

So, after having found the evaporation per pound from and at 212 deg. F., multiply the particular efficiency curve so far used, by the nearest factor of correction, for

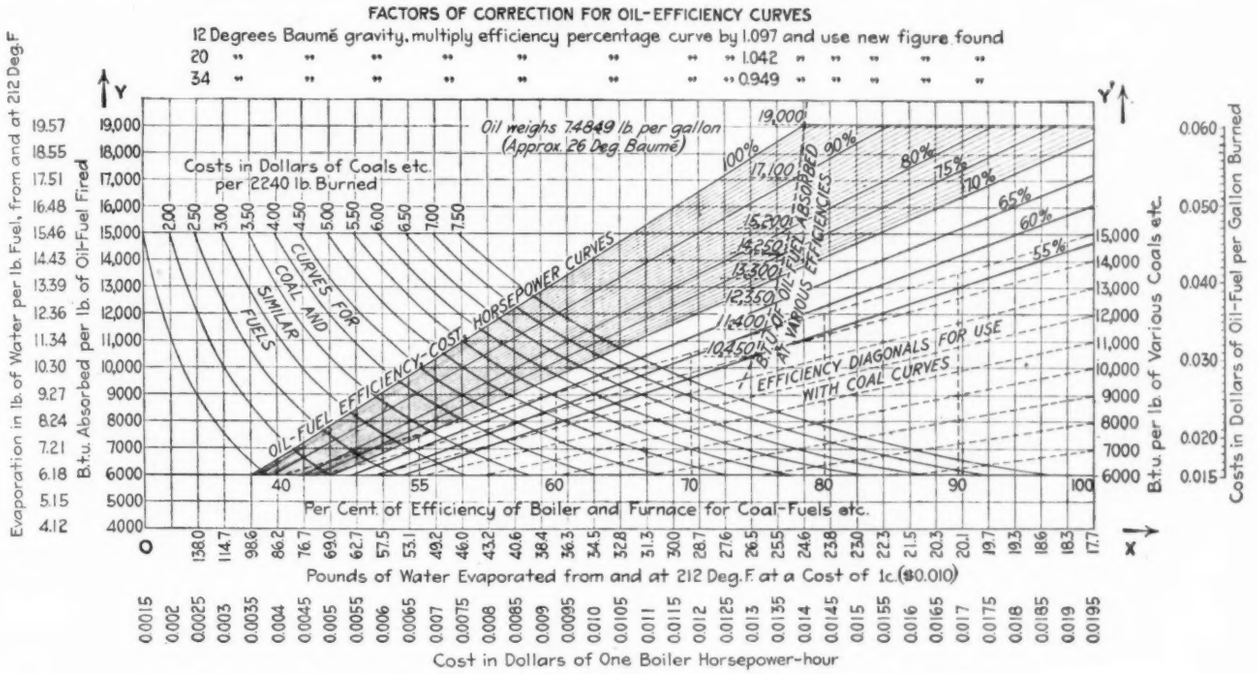

FACTORS OF CORRECTION FOR OIL-EFFICIENCY CURVES
12 Degrees Baumé gravity, multiply efficiency percentage curve by 1.097 and use new figure found
20 " " " " " " " " 1.042 " " " " "
34 " " " " " " " " 0.949 " " " " "
Oil weighs 7.4849 lb. per gallon (Approx. 26 Deg. Baumé)
Costs in Dollars of Coals etc. per 2240 lb. Burned
OIL-FUEL EFFICIENCY-COST-HORSEPOWER CURVES
CURVES FOR COAL AND SIMILAR FUELS
EFFICIENCY DIAGONALS FOR USE WITH COAL CURVES
Evaporation in lb. of Water per lb. Fuel, from and at 212 Deg. F.
B.t.u. Absorbed per lb. of Oil-Fuel Fired
B.t.u. per lb. of Various Coals etc.
Costs in Dollars of Oil-Fuel per Gallon Burned
Per Cent. of Efficiency of Boiler and Furnace for Coal-Fuels etc.
Pounds of Water Evaporated from and at 212 Deg. F. at a Cost of 1c. ($0.010)
Cost in Dollars of One Boiler Horsepower-hour

CHART SHOWING THE RELATIVE COSTS OF COAL AND OIL FUEL

the oil being figured on. (These are given in the center of the chart.) All further calculations should be made in the usual manner on the new efficiency curve thus obtained.

It frequently happens that the data collected at a plant on which to base a comparison show coal ahead of oil fuel from an economic standpoint, where all the signs pointed the other way. If the engineer making the investigation is convinced that the coal figures furnished by the plant engineer represent the true average conditions, then nothing more need be said; but if the "efficiency we are getting with coal" does not take into account banked fires and the low points in the efficiency curve as well as the high spots, then there is only one thing to do, and that is to take observations for a long enough period to insure the accurate determination of the average overall efficiency with coal. Experience with oil fuel then enables the engineer to decide what average overall efficiency can be looked for with liquid fuel at the plant in question. The rest is but a matter of a few moments' use of the chart.

Laying Out an Eccentric Elbow

By M. W. Ward

It is sometimes necessary to join two pipes inclining at two different angles; that is, angle of turn and angle of inclination, as in Fig. 1. To make drilling templates for the flanges the following methods may be used. The example here detailed was one of six different elbows for a job in South America, the purchaser requiring all flanges to be drilled ready for assembly, with a penalty for non-fit. All the flanges were drilled regular straddle center line, and the elbows drilled to special templates determined by the method described, the greatest error being less than $\frac{1}{32}$ in.

Fig. 2 shows the graphical analysis, which, if made to a large scale, will give fairly accurate results. The analytical method is to be preferred on account of accuracy and ability to check the results. The following example was worked to four places of decimals with seven-place logarithms.

Referring to Fig. 2, first construct the triangle ABC, making AB any desired length, say 25 ft. Solve for angle BAC; then

$$\tan BAC = \frac{12.4167}{34.5}$$

$$= 19 \text{ deg. } 50 \text{ min. } 9 \text{ sec.} \quad (1)$$

$$\text{Side } BC = AB \tan BAC = 9.0182 \quad (2)$$

$$\text{Side } AC = \frac{AB}{\cos BAC} = 26.5768 \quad (3)$$

Construct the triangle ACD; the angle DAC is equal to the angle of inclination, namely, 41 deg. See Fig. 1. (Note that the angle of the elbow is always the angle of inclination or rise and is independent of the angle or turn.)

$$\text{Side } DC = AC \tan DAC = 23.1029 \quad (4)$$

Construct the triangle ALH, in which $AL = BC$ and $LH = DC$; solve for angle HAL:

$$\text{Tan } HAL = \frac{LH}{LA} = 68 \text{ deg. } 40 \text{ min. } 36 \text{ sec.} \quad (5)$$

Construct elevation of flange No. 1 (in direction of arrow shown on Fig. 1). See Fig. 3, in which the line HA of the triangle HAL will represent the true center, or parting line of the flange. All that is required is the chord length ab to the required center line of the flange, which equals

$$2\,Aa \sin \tfrac{1}{2}\,aAb = 5\tfrac{39}{64}+ \text{ sec.}$$

Having found the required center line, proceed to lay out the bolt holes, straddling this line in the usual manner.

For flange No. 2 refer to Fig 2, construct the triangle DEC with EC perpendicular to DA. The angle DCE equals angle DAC, which is 41 deg.

$$\text{Side } EC = DC \sin CDE = 17.4360 \quad (1)$$

Construct the triangle AGL, in which the angle ALG = angle BAC, namely 19 deg. 50 min. 9 sec.

$$\text{Side } LG = AL \cos ALG = 8.4831 \quad (2)$$

$$\text{Side } AG = AL \sin ALG = 3.0601 \quad (3)$$

Now produce the line GL to point F, making $GF = DC$; establish point M on the line AD by connecting points FC.

Side $AC = 26.5768$ (see 3 for flange 1); then

$$GC = AC - AG = 23.5167 \quad (4)$$

To solve for angle GCF:

$$\text{Tan } GCF = \frac{FG}{GC} = 44 \text{ deg. } 29 \text{ min. } 29 \text{ sec.} \quad (5)$$

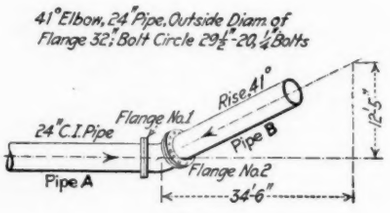


FIG. 1

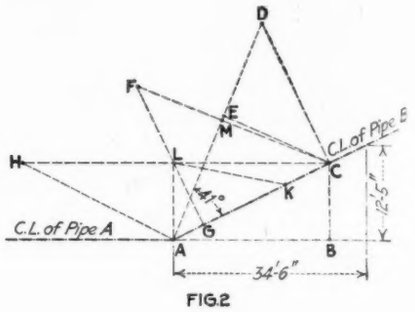


FIG. 2

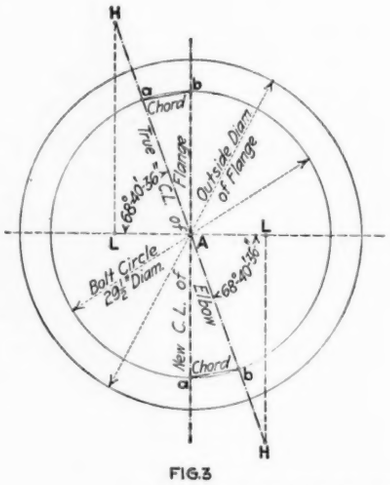


FIG. 3

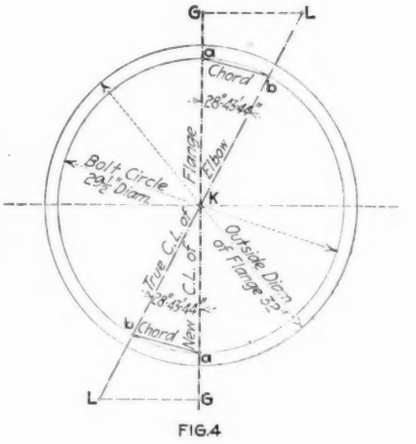


FIG. 4

DIAGRAMS MADE IN DESIGNING A 45-DEG. ELBOW

$$\text{Side } FC = \frac{FG}{\sin GCF} = 32.9664 \quad (6)$$

Angle MCE, 90 deg. minus the sum of the angles DCE and ACF or 4 deg. 30 min. 31 sec. (7)

$$\text{Side } MC = \frac{EC}{\cos MCE} = 17.4901 \quad (8)$$

$$\text{The length } FM = FC - MC = 15.4763 \quad (9)$$

Construct the triangle GLK; the side $GK = FM$. Now to solve for angle GKL:

$$\text{Tan } GKL = \frac{LG}{GK} = 28 \text{ deg. } 43 \text{ min. } 44 \text{ sec.} \quad (10)$$

Construct elevation of flange 2 (in direction of arrow. Fig. 1). See Fig. 4 in which the line LK of triangle GKL will represent the true center or parting line of flange No. 2. Obtain the chord length ab to the required vertical center line of the flange which equals $2\,Ka \sin \frac{1}{2}\,aKb = 7\text{-}21/64$ sec. Construct the template in the same manner as for flange No. 1.

Simple and Convenient File

By S. H. Viall

Often a reader comes upon information that he thinks might be of value to him for future reference. Most men can remember interesting information for a few days, some for a few weeks, but very seldom can they recall from memory with sufficient accuracy all the details when they are really needed. This inability to remember has caused many engineers to adopt schemes for filing information that may be of value to them at some time. Along this line *Power* has published a number of ideas contributed by readers. Some of these plans call for the keeping of all copies of the periodical and having them bound with an index issued by the publisher. Other systems call for card-indexing of the articles and for various schemes of filing the papers. In fact, the ideas are almost as numerous as the individuals that gather the information.

Filing the volumes of a publication necessitates keeping several hundred pages of material that may not be of value and for which there is often a lack of storage space. Besides, there is considerable expense attached to this method. As the years go by, the bulk of the volumes accumulated becomes formidable and the work of locating a certain article of perhaps twenty lines upon a given subject becomes laborious.

The card-indexing plan is well adapted to the plant or office that has a large and elaborate filing system with special clerks to attend to the routine details. To the average engineer, who must himself do all the clerical work connected with such a system, it becomes annoying because he often feels that he has not the time to sit down and make out cards, file them, and then file the articles or magazines.

The writer has used four or five ideas during the past ten years, and from experience has developed a simple plan that requires a minimum of labor to keep in perfect order. This same filing system is well adapted to the needs of the operating engineer, or to any other man who preserves special information that has particular value to his needs.

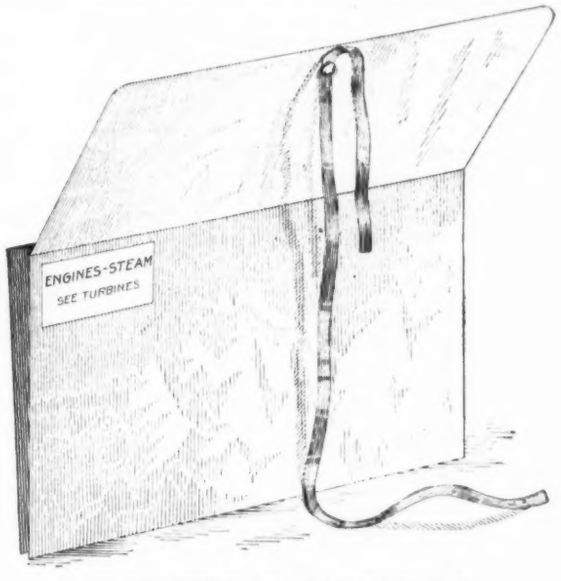
The system makes use of a number of large envelopes of the expansion type and large enough to take flat pages of the papers from which most of the clippings are cut. On one of the upper corners the envelope is labeled to designate the character of information filed as shown in the illustration.

Filing to be by Class, then Subclass

It is advisable to file the articles under the noun designating a general class. Should the material in any one envelope become bulky, it can be separated into subclasses. For example, an envelope is labeled "Meters." In it are filed articles, illustrations, data, etc., cut from the technical papers, manufacturers' circulars or from other material relating to meters, whether they be water meters, electric meters, steam meters, gas meters, or meters for some other purpose. When the envelope is filled some of the material can be removed and two, three or more envelopes used. All will be marked "Meters," with a subhead stating what kind of meters. In like manner one or several envelopes can be labeled "Transmission" and will contain articles on rope drives, belt drives, shaft drives and electrical transmission.

After the envelopes are labeled they are placed in alphabetical order. In going through the files occasionally it is an easy matter to throw away articles that have lost their value and in this way keep down the bulk of the material and only preserve that information which is uptodate.

A fireman may have his file and label the envelopes just the same as the chief engineer, although the two men will undoubtedly select different articles. The fireman will choose from the reading matter or advertisements illustrations that show clearly how valves are made or the interior construction of injectors and other equipment about the boiler room. In fact, he will preserve almost all the good cross-sectional or phantom illustrations that come to hand, as well as such educational articles as he can understand. As the man progresses and gets more responsible work, he may throw away much of this early material because the information contained therein has become well fixed in his mind. The file is so elastic that he can do it readily without rearranging the entire system.



FILING ENVELOPE FOR CLIPPINGS

It has become the custom of the writer to read a technical paper with pencil in hand, preferably a red one, and as he reads, to mark the articles that interest him, also marking sentences, paragraphs, tables, etc., that are of particular interest. The paragraphs in an article so marked in red are conspicuous and are easily located at some future time. The word in the title to the article corresponding to the label of the envelope is also underlined with a red pencil. Or if there is no corresponding word in the title, the index is written on the sheet in red pencil. In this way there is very little chance of filing it in the wrong envelope. If there are several sheets to the article to be filed, it is advisable to fasten them together with O.K. clips. The use of two clips prevents other sheets getting entangled readily, as they otherwise might, and the O.K. clips have the advantage that other papers do not readily catch under them. When fastening the sheets together the title should always be exposed, even though this requires reversing the position of the title sheet. All matter not belonging to the article should be crossed off, so that no time may be lost when again referring to these particular sheets.

Frequently it happens that two articles worthy of preserving will appear on opposite sides of the same sheet. It is usually advisable to select the more valuable article and disregard the other, as it is impossible for any person to save all the interesting information coming to his attention. If it is desirable, however, to keep both articles, file the sheet under either one of two headings. Upon letter paper write the title of the second article, its substance, and under what index the sheet is filed. Either article can then be easily found when needed.

Envelopes can be had at first-class stationary stores at a price of a few cents each, depending upon quality and size. The manila envelope is rather light for frequent handling. Those made of heavier brown or red paper are better. The number will depend upon the individual doing the filing. Some men might do with a dozen for a starter, others will need 25, and those interested in a number of subjects will require 50 or more envelopes.

Anyone starting to preserve information of the character mentioned will be pleased to learn how easy it is to keep up the file after the first work of marking the envelopes. It is certainly worth while to keep such a file. In a comparatively short time it is possible to collect a great deal of valuable information on the various subjects selected, and the great advantage of such a system is that the information is all together in small bulk and is available at any time.

Headings for General Use

The following list of headings used by the writer may serve as a suggestion to others wishing to adopt the system of filing just described.

Air (compressed, lift); Battery (storage); Breechings; Boilers (types, accessories, explosions); Boxes (starting); Cement; Chimneys; Coal; Coal (powdered); Combustion; Condensers (steam); Conveyors (coal and ash); Costs; Definitions; Draft; Drawings; Electrical (wiring, miscellaneous data and apparatus); Elevators; Engines (steam, internal-combustion); Forms (costs, records, reports); Furnaces and Stokers; Generators (electrical); Heaters (feed-water); Heating (systems, data); Illumination (lamps, reflectors); Indicators (steam-engine); Injectors; Instruments (indicating, recording, measuring); Kinks; Laws and Ordinances; Locomobiles; Locomotives; Mathematics; Meters (steam, gas, water); Miscellaneous; Motors (a.-c., d.-c.); Oils (lubricating and cylinder, oiling devices); Packing; Piping (capacity, special installations); Producers (gas); Pumps (steam, power); Questions (examination); Refrigeration (machinery, ammonia); Regulators; Separators; Steam (saturated and superheated); Strainers; Tests (boiler, engine, stoker); Tools; Transmission (rope, belt, shaft, electrical); Traps (steam); Transformers (electrical); Tubes (boiler, condenser); Turbines (steam, water); Valves; Ventilation.

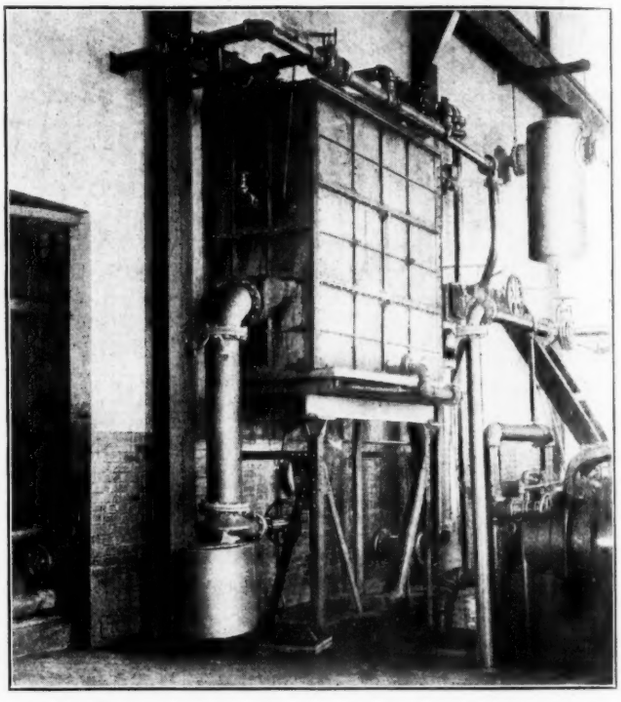
Speed of Minimum Friction is the rate of motion at which, if varied from, either faster or slower, the friction of a bearing increases. With perfect lubrication it is believed to be between 100 and 150 ft. per min.

Coefficient of Friction means the proportion of the pounds pull (or push) to the pounds (weight) being moved over a given horizontal surface; i. e., wood on wood without lubricant may have a coefficient of 0.5, or require ½ lb. pull to move one pound along slowly. Or, coefficient of friction is the ratio of the force required to slide a body along a horizontal plane surface to the weight of the body.

Locating the Feed-Water Heater

There seems to be no rule for the placing of a feed-water heater. Like the feed pump, in the majority of steam plants it is put wherever convenient—above the boiler, in the basement, in the boiler room, in the pump room or in the engine room. In many instances it occupies a prominent position and prevents the placing of other apparatus to advantage.

The accompanying illustration shows how a feed-water heater is placed in one large power plant. The wall di-



HEATER MIDWAY BETWEEN ENGINE AND BOILER ROOMS

vides the engine room from the boiler room. The heater is placed about central in the wall which faces the rear ends of the boilers. It is supported about 6 ft. above the floor by a platform made of angle irons and I-beams.

This arrangement brings a part of the heater in the boiler room, where the space could not be utilized for any other purpose, and elevates the heater out of the way in the engine room.

J-M Insulating Materials

A boiler room ceiled with Transite asbestos lumber is fire-resisting to a high degree, as the lining does not readily transmit heat and consequently the woodwork does not acquire tinder-like dryness. In case of actual fire the flames are not likely to pass the asbestos lining.

Where partitions are pierced for the passage of flues or pipes, an insulation panel is easily inserted. For electric switchboards, switch boxes, partitions and other central-station installations working at high tension, Ebony asbestos wood can be used.

By the judicious use of these two materials in such places as indicated, many old buildings may be protected to a degree against fire. These insulations are the products of the H. W. Johns-Manville Co., of New York.

Editorials

Higher Steam Pressures

After all, human nature is much the same the world over and in every channel of human effort. We all exhibit the "easy come, easy go" characteristic. For example, a tour of inspection of plants in districts where coal is plentiful and cheap, as in many parts of Pennsylvania, will show the average plant to be much lower in economy and the maintenance more neglected than in plants in New England, where a ton of fairly good steam coal brings nearly as much as a barrel of flour even in these turbulent days.

It is some time since the various forms of prime movers became more substantial than ideas, and since then engineers have turned their attention to gaining economy through hitherto neglected sources.

Pressure is one of these. It is a big step from the pumping engines of Newcomen using atmospheric pressure to the two-hundred and twenty-five pounds used in the new Connors Creek Station of the Detroit Edison Company and the two-hundred and twenty-five or fifty common on ocean and railway. But the step has been gradual.

Watt called Trevithick a murderer because he wanted to use fifty or sixty pounds. George Corliss had much to answer for when he advocated the use of one-hundred and fifty pounds; and that was not long ago.

We have foiled the elusive heat unit by buying supplies on specification, by superheat, by vacuum, by getting the last molecule of carbon dioxide and by several other "refinements." And now comes a veritable "hop, skip and jump" in pressure, the advantages and requirements of which were told at the recent meeting of the American Society of Mechanical Engineers and which appear elsewhere in this issue. For some time rumors have indicated a change upward, and now the question comes forth for discussion. The increase to a thousand pounds, of which we have heard, would seem to be unwarranted, as the ideal efficiency does not increase very rapidly above four or five hundred pounds, especially with high degrees of vacuum.

Heating Value of Live and Exhaust Steam

A recent editorial entitled "Does Live-Steam Heating Boost the Coal Bill?" has been misinterpreted into a plea—or an excuse—for the use of live steam instead of exhaust for heating and industrial purposes.

Power has pointed out too many times and in too many ways the fallacy of throwing away heat units in the discharge from a condenser and taking fresh ones from the boiler to do what those discarded might have done, to be suspected of this heresy. Another attempt to illustrate and drive in this fact is made on page 821 of this issue.

But there is another issue between live and exhaust steam. There are those who maintain that a pound of exhaust steam will do more heating than a pound of live steam of the same pressure and temperature. We can see no reason why this should be so and have yet to be convinced that it is so. In saying this we do not discredit the assertion of many reputable engineers that it takes more coal to keep their buildings at a certain temperature when using steam directly from the boiler through a reducing valve than when using exhaust steam. Inasmuch as all the heat that is put into the steam is diffused through the building in either case, it does not seem as though there should be any difference. True, some of the heat units in the case of the exhaust steam have been converted into power, but in almost every case this is reconverted into heat—be it in the electric light or motor, the friction of the bearing, the impact of, or resistance overcome by, the tool—within the building itself. Perhaps it is the failure of a building to take more exhaust in proportion to the amount converted into power which has led to this idea that exhaust steam has a greater value pound for pound than live steam. Perhaps it is the failure to allow for the no inconsiderable amount of heat contributed to a building by its inmates—an average adult is estimated to give off some four or five hundred B.t.u. per hour—and for the fact that it naturally takes more heat to keep an empty idle building warm in the night than it does a teeming busy one in the daytime. We do not know of any tests that have been made, with sufficient care to eliminate all the other variables, which indicate that a pound of steam of given pressure, temperature and quality has any more or less heating value, whether it arrives in the exhaust of an engine or through a reducing valve.

The Lubrication of Cylinders and Bearings

Many of those who read in last week's issue of *Power* the lecture on the lubrication of cylinders and bearings, by F. L. Fairbanks before the Student Branch of the American Society of Mechanical Engineers at the Worcester Polytechnic Institute, were likely impressed by the means adopted to make shutdowns or serious trouble due to lubrication or lack of it extremely remote. The experience of most of us with lubrication includes incidents that make it unnecessary to argue the worth of close study of the problem. The characteristics of the materials of bearings and journals and the properties of initial and fresh charges of oil to the system receive universally thorough attention.

The lecturer points out that if as thorough attention were given to daily checking the specific gravity and viscosity of the oil and to frequently watching the oil temperatures before entering and after leaving the bearing, this thoroughness would give complete and not halfway results. Good filters will slowly but surely take out and hold some of the "fatty" matter in the oil, reducing it in those beneficial materials. Summer tem-

peratures impair the viscosity of the oil, and when it becomes so weak as to be unable to adequately support the journal, the trouble increases and "feeds on itself." This is an excellent explanation of the often baffling cause for serious heating of many bearings at one time. Frequent reports of the gravity and viscosity will show when the oil is wearing out and give opportunity to correct these properties before sudden loads, or increased speeds, or room temperature bring on trouble.

To the designer, the great amount of piston bearing surface and the threading of grooves in the cylinder and the grooving of the piston to make the latter ride on a film of oil are of much interest because of their success, as demonstrated by the lecturer. And journal clearances; how often are these made so as to give a correct margin of safety for temperatures to which the journals may possibly be subjected? Most practical men know that there is a full measure of truth in the lecturer's statement that a ten-inch and a twenty-inch journal of the same material will often be given the same clearance. There are indeed few plants where keying-up is not guesswork the accuracy of which depends upon the operator, or human factor—always an uncertain quantity and varying in the same man at different times.

The keying-up by thousandths of an inch with the aid of wedge screws designed to give so many thousandths per turn, further assisted by tables worked out showing how much clearance to allow for running and how much excess to give for expansion due to possible temperature rises, is a refinement that may seem extreme, but that is worth its cost when it prevents the always costly shutdown—costly not only in dollars and cents to the owners, but in the depreciation of the operator's reputation.

"We Have No Engineer"

It is difficult to realize that although engineers and manufacturers are devoting a good deal of time and spending large sums of money in the endeavor to obtain economy in the steam plant, there are many plants that are being operated in a "just throw in the coal" manner of firing the boilers.

In a small steam plant where the coal consumption does not exceed a ton per day, one frequently finds in charge a man who, although careful, makes no pretense of being an expert on fuel economy but merely operates his small plant the best he knows how. The cost of the yearly supply of coal does not impress the owner as being sufficient to warrant employing a higher-grade engineer. This condition is common, but the number of power plants so operated is gradually becoming less as the owners and managers awake to the fact that the steam plant is the greatest overhead charge connected with the business.

However the boiler-plant operation may appeal to the owners of small plants, one can hardly understand how anyone concerned with the profits of the business can permit the steam end to be operated by inefficient men.

One case in particular is offered as evidence that the matter is of little concern to some owners. This plant is in a manufacturing city of Virginia. It is true that coal is cheap in that section, which is near the coal-producing region, but when it is considered that the boiler plant is close to one-thousand horsepower capacity, the coal item in a year amounts to a tidy sum, even if fuel is comparatively cheap.

This plant has no engineer, according to the statement of the officials of the company. The stoking is done by negro firemen, who undoubtedly know next to nothing regarding economical operation and principles of combustion, and in fact little if anything of modern boiler-room practice. The man in charge of the steam plant is a general "all round" man engaged in this and that occupation and devotes but little if any time to the boiler-room force. If steam is kept to the desired pressure, that is sufficient; the coal pile does not count.

The conditions of the boiler plant are not known. For the sake of an example, assume that it requires four pounds of coal per horsepower. This, with a ten-hour day and three hundred working days per year, would amount to forty thousand pounds, or twenty short tons, of coal per day, or six thousand tons per year. At a cost of, say, two dollars per ton the fuel bill would be forty dollars per day, or twelve thousand dollars per year.

Unless a steam plant is in first-class condition in every respect, the right man can make a saving in coal. In the plant under consideration and burning twenty short tons per day, it would not be much of an achievement to save two tons per day, or four dollars on the fuel bill, an amount equal to twelve hundred dollars per year. That, added to what the man who now has charge of the boiler receives, would pay the salary of a first-class man to devote his entire time and energy to the boiler room.

"We have no engineer" doubtless means nothing to the officials of this company, who in all likelihood do not know much more about boiler operation than their colored firemen. If a common workman in the plant asked for an increase of five cents an hour in wages, he would probably be discharged or forced to work for the old wage, as the manufacturing expenses have to be kept down. Such is inconsistency.

This is one case out of many. There are other manufacturers who "have no engineer." Some of them know it, and some do not. With some this is a case of ignorance, with others it is false economy. Whatever the reason, it is a costly proposition, and a sample of incompetency which would not be tolerated in any other branch of the business.

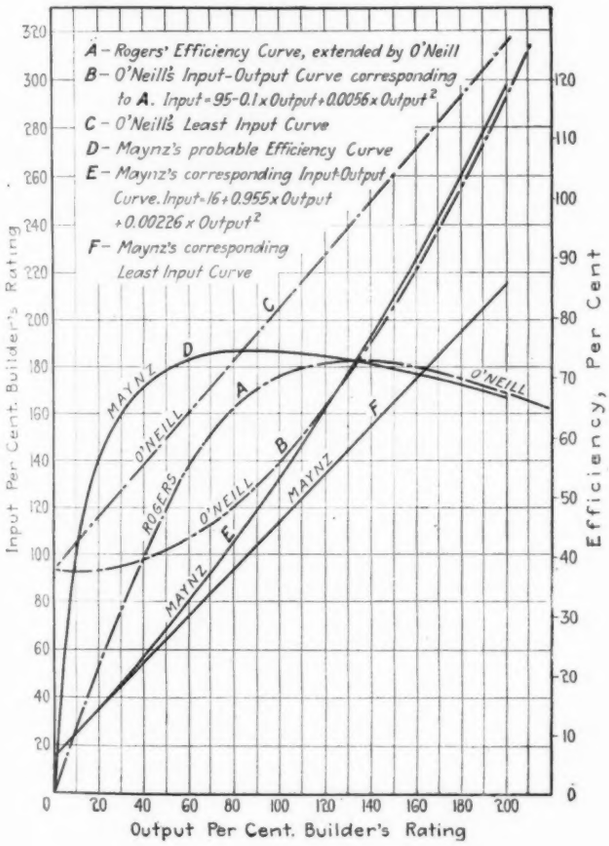
Without detracting at all from the regard in which the many illustrious former presidents of the American Society of Mechanical Engineers were held, none of them, upon retiring from office, received a more whole-souled farewell than that given "Uncle John" last Tuesday night at the annual meeting. Nor was it for his scientific achievements, much as they entitle him to the greatest respect, but because Dr. Brashear is a lively proof of the fact that a man can be a scientist or an engineer or both and still retain his humanity. As he said himself, "Remember me as one who loves his kind, and I don't care what else you think of me."

Is it not singular how unanimously and devoutly the opponents of government protection of the rights of the people in the water powers of the country have become impressed with the holiness and inviolability of the Rights of the Sovereign States?

Correspondence

Boiler Tests at Various Loads

Referring to the articles and letters on the subject of "Boiler Tests at Various Loads," by Messrs. Rogers, Azbe, O'Neill and myself, *Power,* July 20, Sept. 7, Sept. 21 and Oct. 19, I wish to add the following: Mr. O'Neill's mathematical relations of input and output, *Power,* Oct. 19, prove that Mr. Rogers, *Power,* July 20, did not operate the stoker properly at low ratings, as shown by the abnormal no-load losses. The "least possible input curve" by Mr. O'Neill proves the inefficiency of operation at these loads by lying above the actual input curve. I have


A – Rogers' Efficiency Curve, extended by O'Neill
B – O'Neill's Input-Output Curve corresponding to A. Input = 95 + 0.1 x Output + 0.0056 x Output²
C – O'Neill's Least Input Curve
D – Maynz's probable Efficiency Curve
E – Maynz's corresponding Input-Output Curve. Input = 16 + 0.955 x Output + 0.00226 x Output²
F – Maynz's corresponding Least Input Curve
MAYNZ
O'NEILL
ROGERS
Input Per Cent. Builder's Rating
Efficiency, Per Cent
Output Per Cent. Builder's Rating

BOILER EFFICIENCY CURVES

plotted a probable efficiency curve for a boiler stoker fired similarly to Mr. Rogers', showing a maximum efficiency of 75 per cent. at 80 per cent. of rating. This is not an actual test curve, but when compared with Mr. Rogers' curve, which I have replotted together with Mr. O'Neill's analysis and extensions, shows what Mr. Rogers would probably have obtained if the stoker operation during his tests had been more efficient. My curve shows that the no-load loss is about 120 hp., which would give a banked-fire requirement of 11 per cent. of the coal used to produce builder's rating, which is quite normal. The least possible input line lies below the actual input-output curve, whose equation is:

$$Input = 16 + 0.955 \times output + 0.00266 \times output^2$$

An efficiency of 75 per cent. can be obtained with a large Stirling boiler and Murphy stoker, if both are in good condition and properly operated. The ratio of heating surface to grate area of 62.7 : 1 leads an experienced boiler operator to expect the highest efficiency at slightly below builder's rating rather than at 130 per cent. of rating, as Mr. Rogers' extended curve shows.

Mr. Azbe, *Power,* Sept. 21, criticizes my statement that the draft used on the low rating was too high and contends that the draft is varied by the thickness and condition of the fuel bed and the character of the coal. This is correct, but at a given rating there is a certain thickness of fire and a certain corresponding draft that will give the highest efficiency for a certain coal. This point is especially well brought out in the tests made by A. P. Kratz, and published in "A Study of Boiler Losses," Bulletin 78 of the University of Illinois. [See *Power,* Nov. 30, for complete abstract of this bulletin.—Editor.] On pages 18, 19, 30 and 31 of the bulletin there are curves showing the variation in efficiencies with the thickness of fire and the intensity of furnace draft at various loads. The figures that Mr. Azbe gives to show that a boiler developed 500 hp. with 6, 7, 8 and 9 in. thickness of fire and corresponding drafts of 0.17, 0.23, 0.29 and 0.45 in. do not prove much, as he does not state the efficiencies obtained. I do not doubt that one can develop any desired rating with nearly any fire thickness, but one cannot do so efficiently unless there is the correct relation of character of coal, fire thickness and draft. In operating a boiler one first sets the damper to give the draft to burn the quantity of coal required for the rating and then operates the stoker to obtain the correct thickness of fire to give the highest CO_2 (without CO) and the highest efficiency. From this reasoning, together with Mr. O'Neill's analysis, there is no doubt that at the lower outputs Mr. Rogers did not have the proper combination of fire thickness and draft to give the best efficiency at the combustion rates obtained. The variation in draft required to burn different coals efficiently is illustrated by the following from tests run by me:

BOILER TESTS AT VARIOUS LOADS

Test	1	2	3	4	5
Boiler	Aultman & Taylor	Aultman & Taylor	Stirling	Stirling	Oil City
Stoker		Combustion	engineering	type E	
Rated horsepower	394	394	346	346	285
Ratio of heating surface to grate area	57.5 : 1	57.5 : 1	57.6 : 1	57.6 : 1	52 : 1
Builder's rating developed, per cent	101.5	151.8	99.0	98.4	153.7
Combined boiler, stoker and furnace efficiency, per cent	77.4	73.6	76.1	76.3	70.6
Hours' duration	10	10	11	11	10
Dry coal per sq.ft. grate per hr., lb	18.4	28.1	19.1	19.5	26.7
Draft between stoker wind-box and furnace, in.	0.74	1.13	1.93	1.84	3.16
Kind of coal	Southern (Kentucky) nut	Star and slack	Clinchfield (W. Va.) run of mine	Bewley-Darst (W. Va.) nut and slack	Shawmut (Penn.) nut and slack
Analysis:					
B.t.u. dry	13,738	14,000	14,341	14,079	14,170
Volatile matter, per cent	35.23	36.16	30.87	33.40	31.98
Carbon, per cent	55.22	55.99	61.21	61.02	59.42
Ash, per cent	9.55	7.85	7.92	5.58	8.60
Sulphur, per cent	1.42	1.08	0.46	0.47	3.26

In regard to the way I figure the combustible in the ash, I would advise Mr. Azbe that this is very approxi-

mate and of course useless when the ashes dumped are less than the total ash in the coal, determined by analysis. However, if we have 9,400 lb. of ashes dumped and the ash in the coal, by analysis, is 7,000 lb., the conclusion seems to be reasonable that the difference (2,400 lb., or 34 per cent.) is combustible, provided the ashpit was clean at the start of the test. Some of this may be siftings of coal from the stoker, but if this is mixed with the ash, it must be regarded as such and will increase the percentage of combustible in the refuse. Here are the amounts of ash and combustible from two coals:

Total ash in coal by analysis, lb.	1,888	2,825
Total ash and refuse dumped, lb.	2,329	3,445
Combustible by difference in weights, per cent.	23.4	21.9
Combustible by analysis, per cent.	18.26	22.95

My method is a fair approximation, provided not too much ash has been blown over the bridge-wall into the soot chamber.

Furthermore, Mr. Azbe states that a plant should have several small boilers rather than a few large ones. The statement is of course too broad. Every plant must have separate analysis made of its conditions, operating hours, load factor, etc. Large boilers and stokers cost less per horsepower than small ones, in first cost, operating and labor costs and in space requirements. An interesting point about space requirement is shown by considering two plants, both of 2,000 hp., one consisting of eight 250-hp. Babcock & Wilcox boilers, 9 tubes high and 16 tubes wide, set in batteries with 4-ft. aisles. The same plant, using four 500-hp. boilers of the same type, 18 tubes wide and 14 tubes high, set in the same manner, requires only 57 ft. 4 in. from boiler wall to boiler wall as against 109 ft. 4 in. for the 250-hp. boiler plant. As the boilers are the same length through the drums, the floor-area requirements for the 500-hp. plant are only 52 per cent. of those for the 250-hp. installation, and as the 500-hp. boiler is only 37 in. higher than the 250-hp., the cubic content of the 500-hp. plant will be over 40 per cent. less than that of the 250-hp. installation, both having 9-ft. settings.

There are other considerations besides operating efficiency that must be determined in plant design, and therefore one cannot say offhand what a plant will need in regard to the number and size of its boilers.

New York City. THEODORE MAYNZ.

Can Soot Formation Be Prevented?

I have recently been reading much of the available literature on the subject of soot, and I am surprised at the scarcity of data. Many United States Government tests have been made on boilers, and soot is mentioned in them, but that is about all. We are not told how to prevent soot; we are merely told to keep it cleaned off the tubes, that it is a factor of inefficiency, that it is usually black, composed principally of carbon, one of the chief ingredients of smoke, and that the particles are very fine.

I have been unable to learn whether soot can be kept off of tubes by proper firing. I believe it can or that very little need be formed.

In my opinion the problem is much like the gasoline-engine problem. Gasoline itself is a clean liquid. The air that supports combustion is also clean. Yet, when the two are not mixed properly, there will be a deposit of dirty soot on the interior of the combustion chamber. But where the air is proportioned correctly, there is no soot deposit.

To prevent soot deposit on tubes and flues, proportion the air correctly. It is questionable whether the formation of soot can be entirely prevented.

New York City. W. F. SCHAPHORST.

The S.-T. Uniflow Engine

The writer has noted the remarks of A. A. Whitney, appearing in *Power* under date of Nov. 23, concerning the S.-T. Uniflow engine and has also received through the mails several letters criticizing certain points in this design. He has therefore prepared the following paragraphs in the hope of answering these criticisms and explaining such points as may not be completely understood.

The high economy of the uniflow engine lies, in part, in the fact that this engine always keeps a "hot end"; that is, the cylinder head and the piston face are but slightly cooled by the low-pressure exhaust. As far as thermodynamic temperature conditions are concerned, it makes no difference whether the exhaust port is located at the end or middle of the stroke, the steam will exhaust as quickly; but the centrally located port has the advantage that the direction of flow of this exhaust steam is away from the cylinder head and the piston face, instead of against the latter for the entire time, as in the Stumpf engine. The coldest part of the cylinder is the middle portion opposite the exhaust belt, and when exhaust occurs, the piston face is removed one-half the stroke therefrom instead of lying at that point, as is the case in the Stumpf engine.

Let us consider the operations taking place in a cylinder of the Stumpf type. The exhaust port is opened by the piston and at the same rate as the latter is moving, and is closed by the piston on the return stroke. At this point compression begins and the residual steam is compressed in the clearance space sufficiently to reheat these parts. This is to say, that the economy of the uniflow engine lies not only in the fact that the cylinder head and piston face are but slightly cooled during exhaust, but also in the fact that the remaining steam has

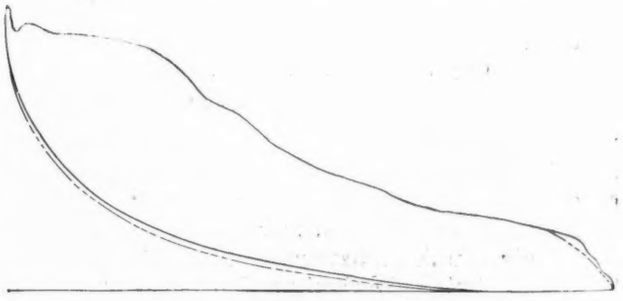


FIG. 1. SAME PORT; NO LEAD

time to be compressed to a point where the heat lost will be returned to these parts. The cylinder walls play a very small part in this operation, but we will consider them later in connection with the sleeve.

Now let us follow these same operations in an engine of the S.-T. type. At the moment of opening the exhaust the piston is moving away from the port at a speed ap-

proximately equal to twice that at which the port is opening. On the return stroke the piston moves an equal distance while the port is closing. In the S.-T. engine the port remains open twice as long as the port in the Stumpf engine, if the ports are the same size, and during half of this time the direction of flow of the exhaust steam has been away from the piston face. Evidently the result of these conditions is that this engine will exhaust approximately twice the amount of steam as the Stumpf engine and without twice the amount of cooling. If the exhaust port were made half the size of the corresponding port in the Stumpf engine, about the same quantity of steam would be exhausted (only the orifice coefficient operating to reduce this amount) with no greater cooling effect. In fact, it is the designer's belief that this cooling effect will be reduced.

We will now consider the cooling effect of the sleeve. The Stumpf engine is nothing more than two single-acting cylinders placed end to end and provided with a common exhaust, each cylinder alternately doing the work. In the S.-T. engine these cylinders have been combined into one, with what amounts to a movable cylinder wall that can be carried from one end of the cylinder to the other and reheated between the times of admis-

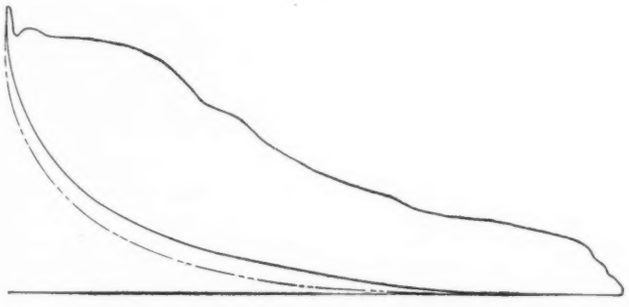


FIG. 2. SAME PORT: ONE-HALF LEAD

sion of live steam. This is the thermal function of the sleeve, and we will examine it more in detail.

In the first place, the amount of cooled metal pushed into the compression end of the cylinder is not as great as the cooled metal in the end of the Stumpf trunk piston. A length of sleeve equal to half the radius of the piston will contain the same amount of metal as the end of the trunk piston (if we consider equal thicknesses). This length, in a correctly proportioned engine, is about equal to half the length of the sleeve, or one-quarter of the stroke. This amount of metal will not be cooled to any greater extent than the same amount of metal in the cylinder wall of the Stumpf engine, and in addition it has the advantage of being moved away from the hot end of the cylinder and up into the clearance end, where it can receive heat from the steam being there compressed. It is the writer's opinion that, after the engine has been running for a short time, this sleeve will acquire a constant temperature.

It should be remembered that the work done in compressing the exhaust steam in any type of engine is not lost, as this work is returned to the piston in heat and actual pressure on the return stroke. The only losses are those due to heat radiation that cannot be entirely prevented.

On clearance the sleeve itself has no effect, except that this clearance may be made a smaller percentage of the stroke, owing to the volume added by the thickness of the sleeve.

As to the effect of the sleeve upon the expansion of the live steam in the cylinders, this effect being due to the difference in the cylinder volume and the volume swept by the piston, it must be remembered that the sleeve is a driving and not a driven element. In fact, the

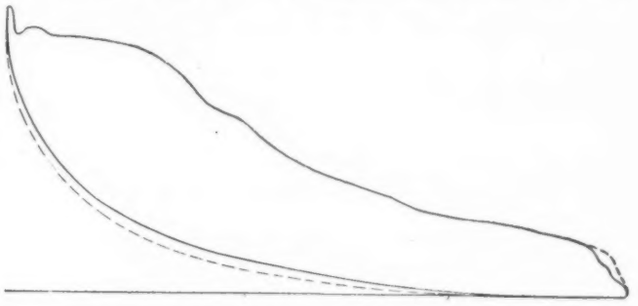


FIG. 3. ONE-HALF PORT: ONE-HALF LEAD

sleeve is, in effect, a hollow trunk piston receiving steam pressure on one end and delivering work at the crankpin to which the connecting-rod from the sleeve is attached. None of the expansive force of the steam is lost, and in this respect an increase in economy is obtained, the result of the more complete expansion of the steam.

The writer himself is not convinced of the economy of giving the piston more than a slight "lead" over the sleeve. Only a steam-rate test can determine this point, as it is not amenable to a purely mathematical investigation. However, a close study of the drawings will show that this lead can be made any amount desired without changing the length of the cylinder. The amount of clearance will increase with the lead.

The cylinder clearance may be made any amount, the only factor affecting it being the compression desired. The writer favors small clearance and high compression as being most economical.

It is not necessary to use a counterbore, as the piston can overrun the end of the sleeve a sufficient amount to keep the wear uniform, but if desired the sleeve can be so constructed. In this regard it is to be noted that the friction between any two parts within the cylinder of this engine is but half the amount of cylinder friction in any other engine, with the result that the parts will run twice as long before there is any necessity for reboring or compensating for wear.

The S.-T. engine has been criticized from the standpoint of leakage. There is no more chance for leakage around the sleeve than around the trunk piston of the Stumpf engine. In fact, the sleeve, as has been before stated, is simply a hollow trunk piston and constructed, as far as packing rings and external finish are concerned, in the same manner as such a piston. Of course there is the combined leakage of the trunk piston and the disk piston to contend with, but the writer does not consider this point as well taken, since neither piston affects the other.

In the accompanying figures the Stumpf engine and the S.-T. engine are compared from the standpoint of their indicator diagrams. In each figure the full-line diagram is that actually made by an Ames-Stumpf Uniflow engine. The dotted lines show the changes in a theoretical diagram from an S.-T. engine of the same size.

The dotted lines in Fig. 1 show the modification of the diagram which may be expected from an engine with exhaust ports the same size as those of the Stumpf engine and with no piston lead. Fig. 2 is the same as Fig. 1, but with the piston given a lead of one-half the width of the exhaust port. Fig. 3 is supposed to be from the same engine with exhaust ports one-half the size of those of the Stumpf engine and with a lead one-half the width of the exhaust port. With an exhaust port one-half the size as that of the Stumpf engine and no lead, the Stumpf and diagrams would be identical.

Rochester, N. Y. H. A. STRINGFELLOW.

Leaving Boilers at Night

In the Nov. 23 issue John Crettol asks if it is dangerous to leave a boiler (presumably under pressure) with a heavily banked fire without attendance at night.

It is, under some conditions. An explosion is not probable if the safety valve is in working condition. If the safety valve or valves are not in good order the danger exists, attendants or not. If carelessness is allowed the dampers may be imperfectly closed when the day attendant leaves, or some unforeseen cause may be the means of opening an ashpit door or damper, whereby steam may be rapidly raised. The foregoing applies only to explosions, but there is always the danger of some pipe or fitting giving way, as in the case cited, which of course can not be foreseen in every case. JAMES E. NOBLE.

Toronto, Ont.

Heating-Season Suggestions

This is the season when most of us give a lot of attention to the heating apparatus in our charge.

I suppose it is true—in fact, we know it must be—that it will take the same amount of steam to heat a building whether it be live or exhaust, but it always seems to me that I get better results from the latter. Perhaps the pulsation of the exhaust has something to do with it.

As a rule, reducing valves are too big. This allows a very slight lift, which in turn quickly cuts out the valve.

In threading pipe the action of the dies breaks up the structure of the metal, leaving it in a good condition to be acted upon by the distilled water flowing through it. If some wise man would get up a pipe joint that required no threading or had the threads rolled on while the pipe was hot, our heating-pipe joints would last much longer.

The heating-system return pipes in our shops were first inclosed in wooden boxes packed with mineral wool. These lasted only a season or two. We have since placed about all of them in tunnels and cement ducts and have fewer repairs to make. We can get into the tunnel to do the work or can withdraw the pipe from the duct and replace it with a new one.

It seems to me that in office buildings where the steam pipes are buried in the floors and walls someone has a lot of trouble in store, for in a short time they are destroyed by the action of the distilled water.

Ours is a common gravity-return system, but it has ample vents at the receiver for expelling the air, and I don't believe any vacuum system requires less attention in operation. We operate it on 1 to 2 lb. back pressure, so I imagine we are not so badly off on the score of economy. Except in the offices, we use wall coils of 1¼-in. pipe with a common globe valve on the inlet and a swing check on the outlet—never was I able to get a factory operative to close an outlet valve, hence the swing checks. J. O. BENEFIEL.

Anderson, Ind.

Collector Rings Rough

Sparkless collection of current from collector rings by means of brushes would seem to be a simpler proposition than obtaining sparkless commutation from a continuous-current machine. Possibly it is so much simpler that its modest requirements are often overlooked. Assuming brushes of the right consistency, sparkless collection at normal loads is largely a matter of good brush contact, of sufficient number of brushes, of correct tension and of smooth collector rings that run true. It must be borne in mind that the speeds of many of the machines on which collector rings are used are high; therefore, nonuniformities incident to slight eccentricities or to slight flats, which would probably cause no trouble on low-speed machines, may cause considerable at the high speeds.

In a certain substation there was a large synchronous converter, the brushes of which wore rapidly and sparked considerably. Even at light loads, when the station was darkened at night a film of light could be seen between the brushes and the collector rings. Similarly rated machines operating on the same duty in the same station and also in other stations showed neither of these symptoms. Smoothing of the rings with a sandstone cut to their curvature seemed to help matters, but only temporarily. Finally it was noticed by an attendant, who happened to be standing where the light from an arc lamp struck the rings just right, that all of the rings had transverse crests and troughs, which suggested a multiplied reproduction of the gear-tooth pitch of some machine in which the collector or the rotor might have been handled.

On taking a light cut off the rings and smoothing them with a stone, all sparking and excessive brush wear ceased.

Schenectady, N. Y. E. C. PARHAM.

Pump Piston Rods Give Trouble

In an office building we had two tandem compound elevator pumps of 900 gal. per min. capacity against 185 lb. pressure. The low-pressure piston rod was screwed into the follower plate on the high-pressure piston, and the high-pressure piston rods and the pump-plunger rods were joined by a cast-iron crosshead, or clamp, made in two pieces and clamped together on the rods by four bolts. Soon after the pumps were started, they gave trouble by the threads breaking at both of these points.

The cast-iron crosshead clamps were replaced by steel, and the follower plates were bored out and threaded for a heavy steel ring, or bushing, which was screwed into the follower and pinned to keep it from working loose. The rod was then screwed in, with a heavy locknut on it. No further trouble has been experienced with these pumps.

Lowell, Mass. R. E. RUGG.

Inquiries of General Interest

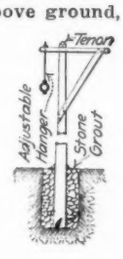
Chimney-Overturning Wind Moment—What would be the overturning moment of the wind which should be provided against in construction of a round chimney 175 ft. high, having an outside diameter of 16 ft. at the base and 9 ft. at the top?
G. F. W.

In most localities it is safe to provide against a wind pressure of 50 lb. per sq.ft. The average diameter of the chimney would be 12.5 ft., and the projected area exposed to the wind would be $12.5 \times 175 = 2{,}187.5$ sq.ft., and the center of pressure, that is, the center of gravity of the projected area, would be about 76 ft. above the base. The effective pressure would be about 25 lb. per sq.ft., and as the total effective pressure would be $2{,}187.5 \times 25 = 54{,}687.5$ lb., the overturning moment of the wind would be $54{,}687.5 \times 76 = 4{,}156{,}250$ lb.-ft.

Gas Equivalent to a Barrel of Oil—How many cubic feet of natural gas will have a heat value equivalent to that of a 42-gal. barrel of California crude oil of 15 deg. Baumé?
O. C. L.

The heat value of natural gas varies according to the locality and ranges from about 700,000 to 1,150,000 B.t.u. per thousand cubic feet at atmospheric pressure, and the average heat value is about 900 B.t.u. per cu.ft. A gravity of 15 Baumé would be equal to a specific gravity of 0.966 that of water, and as a gallon of water weighs 8.33 lb., the weight of a gallon of the oil would be $8.33 \times 0.966 = 8.04678$ lb. and 42 gal. would weigh 337.965 lb. California crude oil 15 deg. Baumé has been quoted as having a heat value of 18,589 B.t.u. per lb., and accordingly the heat value of a barrel of the oil would be $337.965 \times 18{,}589 = 6{,}282{,}431$ B.t.u., and this divided by 900 gives 6,980 cu.ft. of average gas as equal to one barrel of the oil.

Support for Overhead Pipe Line—What is a good method of supporting a 3-in. steam line extended 12 ft. above ground, for a distance of about 200 ft. from a boiler house to a branch department? R. C.



An inexpensive method is to employ a single line of supporting posts with the pipe suspended from crossarms. For a 3-in. steam line the posts could be similar to the sketch and spaced at about 11 ft. centers. Should they obstruct passageways over which a greater span is desirable, the span can be doubled and the middle of the span can be supported by a pair of truss rods.

Kilowatt-hours per Pound of Fuel Oil—In using fuel oil for generation of steam for a direct-connected reciprocating engine, what determines the number of kilowatt-hours developed per pound of oil burned? E. C. S.

The kilowatt-hours per pound of fuel oil will depend on the heat value of the oil, the perfection of the combustion and efficiency of the boiler, the economy and mechanical efficiency of the engine and the efficiency of the generator. Allowing an evaporation of 11 lb. of water into dry saturated steam per pound of fuel oil, 95 per cent. mechanical efficiency of engine, 90 per cent. efficiency of generator and a steam consumption of 20 lb. (weight) of steam per hour per indicated horsepower, then there would be

$$\left(11 \div \frac{20}{0.746}\right) \times 0.95 \times 0.90 = 0.3508 \text{ kw.-hr. per lb. of fuel oil}$$

or as usually expressed, there would be

$$\frac{1}{0.3508} = 2.85 \text{ lb. of fuel required per kw.-hr.}$$

Action of Helical Springs—What are the relative advantages of employing helical springs in compression or in elongation for measurement of force or pressure, as in indicators and dynamometers? G. J. R.

Helical springs used in compression have a tendency to buckle, though errors introduced from this cause can be practically eliminated by appropriate proportions. One advantage of employing springs in compression is that they are protected from overstraining beyond the stress required to close the helix. Springs used in elongation, when provided with axial bearings, are pulled out in a straight line without introducing errors due to buckling, and their change of length is therefore more nearly in proportion to the load or pressure. They have the disadvantages, however, of being exposed to over-elongation from excessive stresses unless protected by a guard or stop to prevent over-reaching. In elongating a helical spring it tends to unwind, and the reverse is true when it is compressed, and with either type, unless freedom of twisting motion is provided for, there will be an irregularity of the elongation or compression from variable frictional resistance at the bearings of the spring.

Unbalanced Weight of Prony Brake—In using a Prony friction brake for ascertaining the power developed by a motor, what is meant by the unbalanced weight of the brake, how is it determined, and how is its value employed in the formula, $HP = \frac{L \times 2\pi \times W \times N}{33{,}000}$? J. R., JR.

The term "unbalanced weight of brake" refers to the force or pressure by which the reading of the weighing scales or weight employed for resisting the effort of the motor is too large or too small by reason of the center of gravity of the brake not being in the vertical plane passing through the axis of rotation of the brake wheel.

For finding the value of the unbalanced weight, loosen the brake clamp from the wheel sufficiently for it to clear the wheel when the clamp end of the brake is supported on a three-cornered file or other knife-edge placed on the rim of the wheel, directly over its center of rotation. The value of the unbalanced weight will then be the weight of the arm when taken at the same point that the effort of the motor is to be weighed.

When, from its direction of rotation, the effort exerted by the motor is assisted by the unbalanced weight of the brake, then, to obtain the proper value of W in the formula, the unbalanced weight is to be deducted from the weight readings obtained in operation of the brake, but when the unbalanced weight opposes the effort of the motor, the value of the unbalanced weight is to be added.

Cooling Water Required for Jet Condenser—What weight of cooling water must be supplied to a jet condenser for condensing one pound of steam under the following conditions: Barometer, 29.7 in.; vacuum, 26 in.; temperature of cooling injection water, 70 deg. F.? S. S. J.

It is sufficiently accurate for most practical purposes to assume the exhaust steam entering the condenser to be dry and saturated and its heat to correspond to the pressure in the condenser and compute the weight of cooling water by the formula,

$$W = \frac{H - t_2 + 32}{t_2 - t_0}$$

in which

H = Total heat of the steam at condenser pressure;
t_2 = Temperature of the discharge water;
t_0 = Initial temperature of the cooling water;
W = Weight of cooling water in pounds necessary to condense and cool one pound of steam to the required discharge temperature.

The temperature of steam corresponding to an absolute pressure of $29.7 - 26 = 3.7$ in. of mercury is found from the steam tables to be about 123 deg. F. But the discharge temperature is less than this, as the condenser pressure is due in part to air carried over with the circulating water and condensed steam, so that the discharge temperature will be 10 to 15 deg. lower than that corresponding to the vacuum shown by the gage. Assuming it to be 15 deg. lower, the actual discharge temperature may be considered as $123 - 15 = 108$ deg. F. The total heat corresponding to a pressure of 3.7 in. of mercury is given by the steam tables as about 1,113 B.t.u. above 32 deg. F. Then as $t_2 = 108$ deg. F., and $t_0 = 70$ deg. F., by substitution the formula becomes

$$W = \frac{1{,}113 - 108 + 32}{108 - 70} = 27.29 \text{ lb.}$$

of cooling water per pound of steam condensed.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Insuring the Coal Supply--I*

By Henry J. Edsall†

SYNOPSIS—The storage of coal is insurance against either excessive fuel prices or a shutdown. Certain precautions must be observed in stocking bituminous coal, and the means to be employed will vary widely with circumstances, each installation being a separate problem. Many varieties of stocking and reclaiming apparatus are here described.

While it has become almost universal practice among business men to carry insurance against fire, accidents to employees and other possible happenings beyond their definite control, the carrying of an ample coal supply as an insurance against delays or a possible closing down of the plant is apt to be neglected by the coal consumer. The advisability of this kind of insurance is not hammered at him by the insurance man as are the other kinds, and he is apt to have a feeling that he can always get coal somehow, even though there is a shortage.

Every once in a while, however, there is a strike of the coal miners, congested conditions on the railroads, an unusually severe snowstorm, an especially heavy demand for coal, along with a business boom and a car shortage, or something else happens to disturb the equilibrium between the available coal supply and the demand. At such times the prices soar, and the man with a good-sized coal pile pats himself on the back for his foresight, while the one who neglected this precaution is bidding against other consumers in the same predicament, for enough coal to keep his plant running and to avoid an interruption to business at just the time when business is booming and any delay would be disastrous.

The consumer with ample storage facilities is also in a position to take advantage of special prices which may be offered him at certain seasons of the year or of a special offer of some dealer because of a combination of circumstances affecting him. Better coal can also be obtained at certain times, and the main supply can be laid in under favorable weather conditions instead of having to handle coal all through the winter, with its snow and sleet storms and zero weather.

Where small anthracite steam coal is used, the storage problem is comparatively simple, since the small size makes it easy to handle by machinery, and it can be piled to any depth without danger of spontaneous combustion. Since however, the majority of the coal used for steam purposes is bituminous, it is this kind as a rule for which storage facilities have to be provided.

BUREAU OF MINES' INVESTIGATIONS

The United States Bureau of Mines has made some investigations of the deterioration and spontaneous heating of coal in storage and expects to carry on further experiments and publish further data later on. Technical Paper No. 16, published in 1912, gives points of information obtained from individuals who have had experience in storing coal under various conditions. It also gives results of tests made by storing different kinds of coal, both exposed to the weather and submerged under fresh and salt water in different localities. The following quotations are from this paper:

The results show in the case of the New River coal a loss of less than 1 per cent. of calorific value in one year by weathering in the open. In two years the greatest loss was at Key West, 1.85 per cent. There was practically no loss at all in the submerged samples in one year, fresh or salt water serving equally well to "preserve the virtues" of the coal. There was almost no slacking of lumps in the run-of-mine samples. In all tests the crushed coal deteriorated more rapidly than the run-of-mine.

The Pocahontas run-of-mine coal in a 120-ton pile on the Isthmus of Panama lost during one year's outdoor weathering less than 0.4 per cent. of its heating value and showed little slacking of lumps.

Gas coal during one year's outdoor exposure suffered practically no loss of calorific value, measurable by the calorimetric method used, not even in the coal forming the top 6-in. layer in the bins.

Submerged storage is an absolute preventive of spontaneous combustion, and on that account alone it may be justified when the coal is particularly dangerous to store and when large quantities are to be stored; but unless the storage period is to be longer than one year, there seems to be no ground for storing any coal under water merely for the sake of the saving in calorific value to be obtained by the avoidance of weathering.

Losses of value from spontaneous heating are a much more serious matter than the deterioration of coal at ordinary temperatures. Oxidation proceeds more rapidly as the temperature rises. The oxidation, beginning at ordinary temperatures, attacking the surfaces of particles and developing heat is probably in some degree an absorption of oxygen by the unsaturated chemical compounds in the coal substance. In a small pile of coal this slowly developed heat can be readily dissipated by convection and radiation, and very little rise in temperature results. If the dissipation of the heat is restricted, however, as in a large pile densely packed, the temperature within the pile rises continuously. The rate of oxidation of the coal platted against the temperature makes a curve that rises with great rapidity. When the storage conditions are such as to allow warming of the coal to a temperature of about 100 deg. C., the rate of oxidation becomes so great that the heat developed in a given time ordinarily exceeds the heat dissipated, and the temperature rises until, if the air supply is adequate, the coal takes fire. Evidently, therefore, it is important to guard against even moderate heating of the coal, either spontaneous or from an external source. Increased loss of volatile matter and of heating value occurs with a moderate rise of temperature, even though the ignition point is not reached.

Spontaneous combustion is brought about by slow oxidation in an air supply sufficient to support oxidation, but insufficient to carry away all the heat formed. The area of surface exposed to oxidation by a given mass of any one coal determines largely the amount of oxidation that takes place in the mass; it depends on the size of the particles and increases rapidly as the fineness approaches that of dust. Dust is therefore dangerous, particularly if it is mixed with lump coal of such a size that the interstices permit the flow of a moderate amount of air to the interior. Coal differs widely in friability; that is, in the proportion of dust that is produced under like conditions. In comparative tests samples of Pocahontas, New River and Cambria County (Pennsylvania) coals produced nearly twice as much dust (coal through a ⅛-in. screen) as coal mined from the Pittsburgh bed in Allegheny County, Pennsylvania. This variation in friability is a factor in affecting the liability to spontaneous heating.

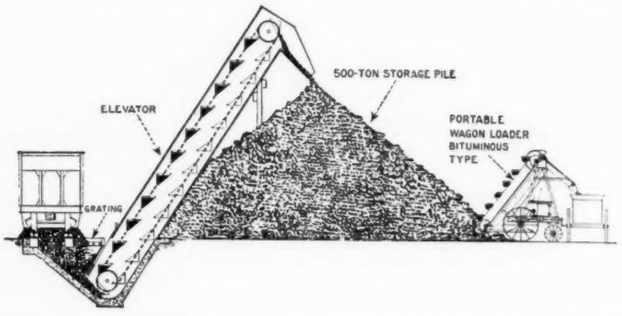


FIG. 1. HANDLING FROM COAL CAR TO CART THROUGH BUCKET ELEVATOR AND PORTABLE WAGON LOADER

Ideal conditions for such heating are offered by a mixture of lump and fine coal, such as run-of-mine with a large percentage of dust, piled so that a small supply of air is admitted to the interior of the pile.

High-volatile matter does not of itself increase the liability of coal to spontaneous heating. A letter of inquiry sent by the bureau to more than 2,000 large coal consumers in the United States brought 1,200 replies. Of the replies 260 reported instances of spontaneous combustion, and 220 of the 260 gave the name of the coal. The 220 instances were distributed as follows: 95 were in semibituminous low-volatile coals of the Appalachian region, 70 in higher-volatile coals of the same region and 55 in Western and Middle-Western coals.

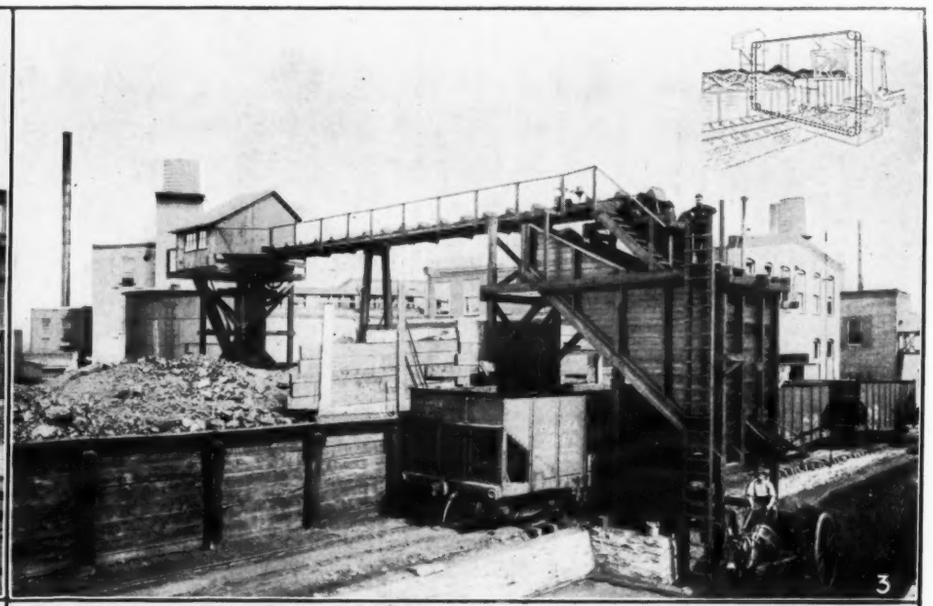
Freshly mined coal and the fresh surfaces exposed by crushing lumps exhibit a remarkable avidity for oxygen, but after a time the surfaces become coated with oxidized material, "seasoned" as it were, so that the action of the air becomes much less vigorous. In practice, coal that has been stored for six weeks or two months and has even become already somewhat heated, if rehandled and thoroughly cooled by the air, seldom heats spontaneously again.

Where mechanical stokers are used, it is almost universal practice to provide an overhead storage bin for at least part of the coal supply, this bin being located either right in the boiler room or at a convenient point close by. This coal is either spouted direct to the stoker magazines or drawn from the bin into a traveling larry or weighing hopper running on an overhead track above the stokers, the coal being delivered from the larry to the magazines.

Where the firing is done by hand, the coal is shoveled from the floor or from a charging car, so that the overhead bin does not show such a decided advantage over a ground-

*Reprinted from "Coal Age," Nov. 6, 1915.

†Link-Belt Co., Philadelphia, Penn.



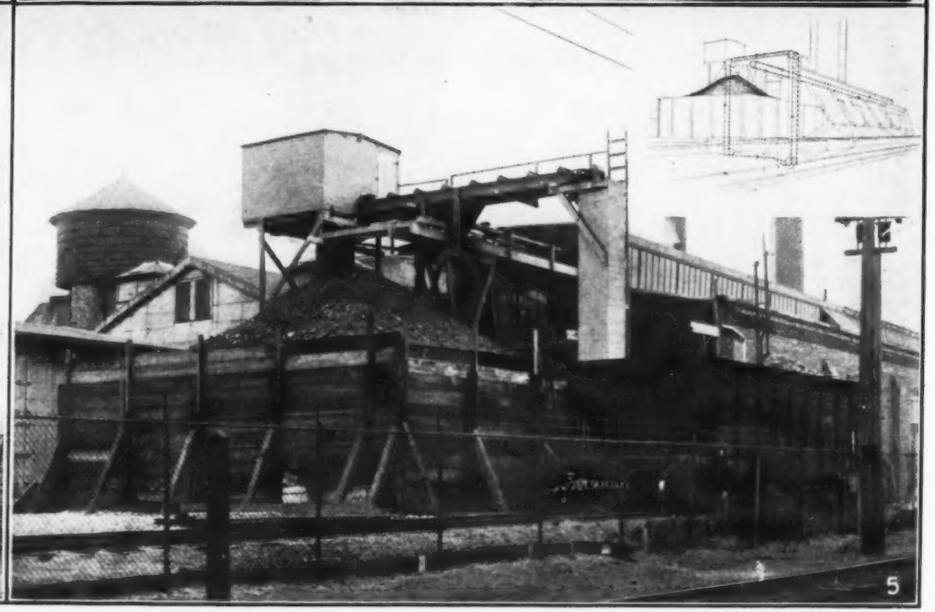


FIG. 2. INCLINED FLIGHT CONVEYOR HANDLING FROM CARS TO STORAGE
FIG. 4. COMBINED BUCKET ELEVATING AND FLIGHT DISTRIBUTING CONVEYORS

FIG. 3. ENCIRCLING CONVEYOR TO TAKE FROM OR DELIVER TO STORAGE
FIG. 5. FLIGHT CONVEYOR AND GRAVITY DISCHARGE MACHINE COMBINED

storage bin. The overhead bin is often used, however, for the sake of neatness and on account of the possibility of arranging it so that the greater amount of the coal can be spouted by gravity to convenient points within easy reach of the fireman without having to keep a lot of coal on the boiler-room floor or constantly bringing it in from the outside.

Without the overhead bin it is customary either to pile the coal against the outside of the boiler-room wall, in front of the boilers, and leave openings in the wall so that it will flow through onto the floor within easy reach of the fireman, or to pile it at as convenient a point as possible for transfer to the boiler room.

On account of the expense of construction the amount of overhead storage is usually kept down to what is considered a minimum for satisfactory operation, and some cheaper type of storage is arranged for the reserve supply, with a means of transfer from the reserve storage to the active storage.

The justifiable amount of investment for reserve storage and transfer facilities depends on the size of the storage that is considered advisable, on the amount of coal which will be delivered to the storage and transferred from it in a certain time and on local conditions. In some cases coal is as a rule received regularly, and the reserve supply is simply an insurance against unusual contingencies, to be called on only at infrequent intervals. In such cases it may pay better to use hand labor partly or entirely for the storage and transference rather than to invest much money to eliminate this hand labor when it does not figure as a large yearly expense. In other cases the coal supply may be so irregular that it is necessary to call on the reserve storage at frequent intervals, and the amount of hand labor which it is possible to eliminate justifies quite a large investment.

If the coal is received as run-of-mine and is to be used in mechanical stokers, it is usually passed through a crusher before it is delivered to the stokers. At small plants it is sometimes passed through a grating over a feeding hopper or the large lumps are broken up in some other way, instead of the expense being incurred of installing a crusher with its driving arrangement. Where the coal is to be fired by hand the crushing is not considered necessary as a rule; but if it is to be handled by machinery, it must be crushed or the machinery must be made capable of handling the largest lumps.

When a crusher is used it can be placed at the receiving point, so that the coal is passed through it as soon as it is unloaded and before storing. On the other hand, if a reserve storage is employed, the coal may be stocked as run-of-mine and passed through a crusher just before it goes into the active storage for the boiler room.

This means collecting from the reserve storage and bringing all coal to the fixed point where the crusher is located, unless this machine is made to move along on wheels, as is sometimes done. Where the coal is handled entirely by conveyors, it simplifies the problem to crush it first, so that it is easier to handle, though this necessitates the storing of crushed coal in the reserve supply as well as in the active storage.

Where a grab-bucket arrangement is used for the reserve supply, it is usually just as easy and sometimes easier to make this a run-of-mine storage and crush the coal just before it goes into the supply for the boiler room. Since the coal mines are fast becoming equipped with screening and picking arrangements for sizing and preparing the coal, it is now entirely feasible to buy coal of the size required for use, though the sized coal costs a little more than the run-of-mine.

In considering the different types of reserve-storage plants the simplest case is undoubtedly that where a moderate amount is unloaded from railroad cars and piled on the adjacent ground without any provision for reclaiming mechanically. The most economical machine for this work is a chain-and-bucket elevator or an inclined scraper-conveyor, technically known as a flight conveyor. A pit is built under the railroad track at the unloading point so that the coal can be discharged from hopper-bottomed cars into the pit and fed through a gate to the elevator or conveyor. With the elevator the buckets pick the coal up, elevate it and discharge it at the head to a chute to the pile. With the inclined conveyor the flights scrape the coal up in a steel trough and discharge it in a similar manner at the head.

The coal can be handled as run-of-mine or a grating can be placed over the track hopper and the larger lumps broken on it, thereby making it possible to use a somewhat smaller and cheaper equipment. For a pile of 500 or 600 tons, the machinery, including a motor for driving, can be purchased for about $600; and in addition to this there will be

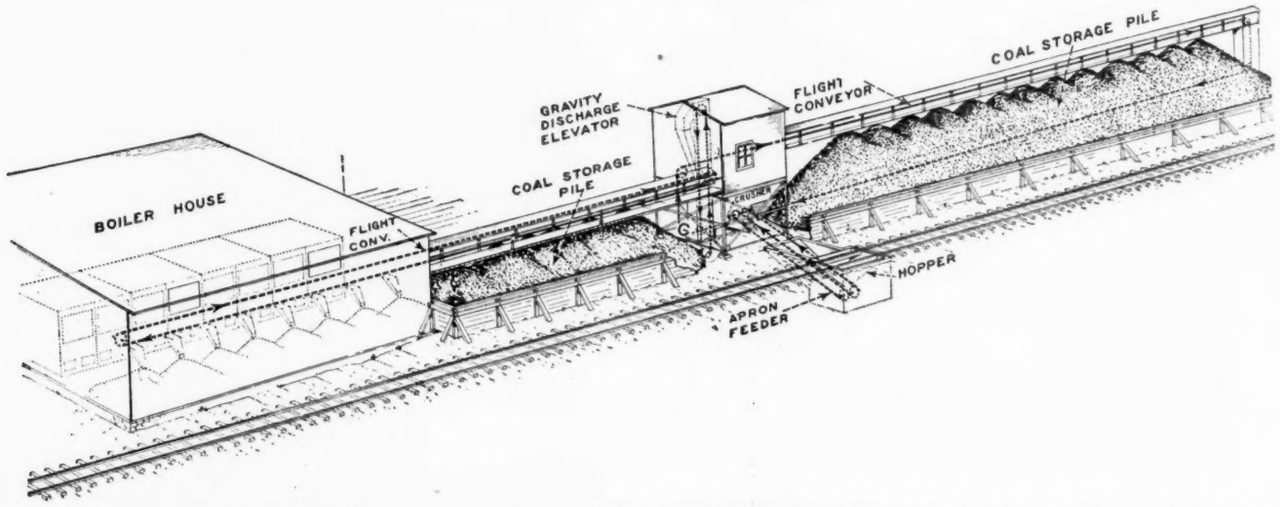


FIG. 6. FLIGHT CONVEYOR DISTRIBUTORS SERVED BY APRON FEEDER AND GRAVITY DISCHARGE ELEVATOR

the cost of the pit and machinery supports, also the cost of installing the equipment. Machines of this kind usually handle coal at the rate of about 30 or 40 tons per hr., providing the coal can be unloaded from the cars at this rate. This depends on the type of railroad car, the percentage of lumps in the coal and the number of men used to unload the cars.

Where a cart is available for hauling the coal to the boiler room or where it can be transferred by small industrial cars, a portable wagon loader is an excellent way to pick it up from the ground. A standard machine of this type will handle anthracite or run-of-mine bituminous coal at the rate of a ton a minute. The machines are built both hand- and power-propelled; the hand-propelled ones cost about $550 and the power-propelled ones $850, complete with electric motor or gasoline engine for operation.

Fig. 1 shows a bucket elevator delivering to a ground-storage pile and a portable wagon loader handling the coal from this storage to carts. Fig. 2 shows an inclined flight conveyor handling coal from the railroad cars to a ground-storage pile, the pile in this case being adjacent to the boiler room so that part of the coal flows through openings in the boiler-room wall to the floor directly in front of the boilers.

By going a step farther and adding a reclaiming feature, a conveyor can be made to deliver the coal to the storage, also take it from the storage and deliver it at a desired point. For this work a gravity-discharge type of machine is often used, because it will both elevate the coal and convey it along horizontally or at an incline, and by bringing the buckets back in a tunnel underneath the pile, the same machine can be used as a reclaiming conveyor. These machines have V-shaped buckets carried between two chains and properly attached to them. When elevating, the coal is carried up in the buckets. When it is being conveyed, the buckets act like

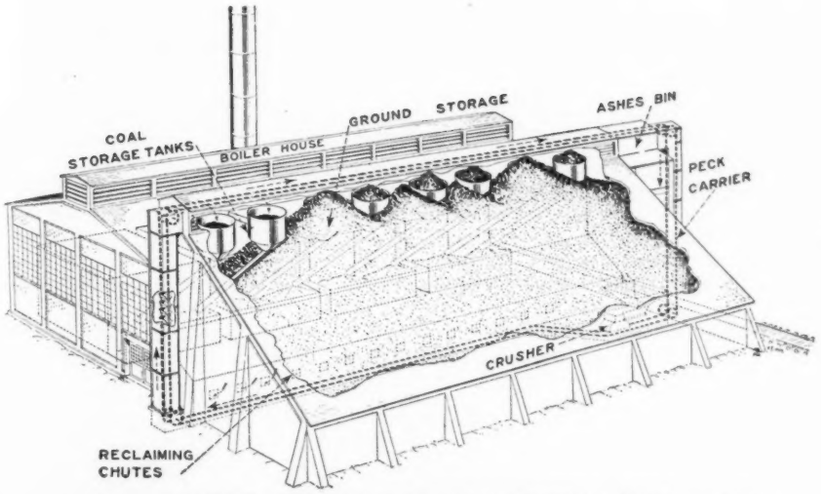


FIG. 8. PIVOTED BUCKET CARRIER CONVEYOR USED WITH RESERVE STORAGE

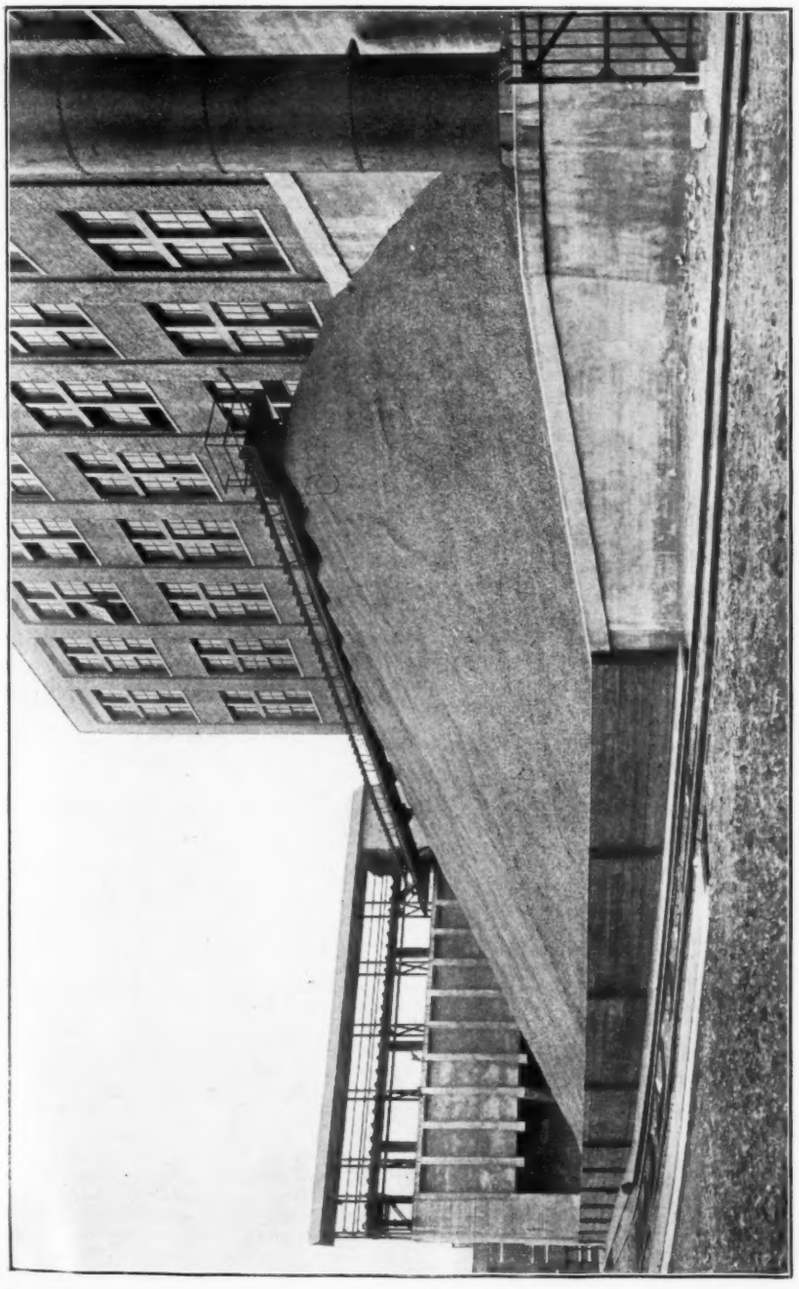


FIG. 7. COMBINATION STORAGE SYSTEM FOR SMALL ANTHRACITE COAL WITH EXCAVATED PILE SPACE

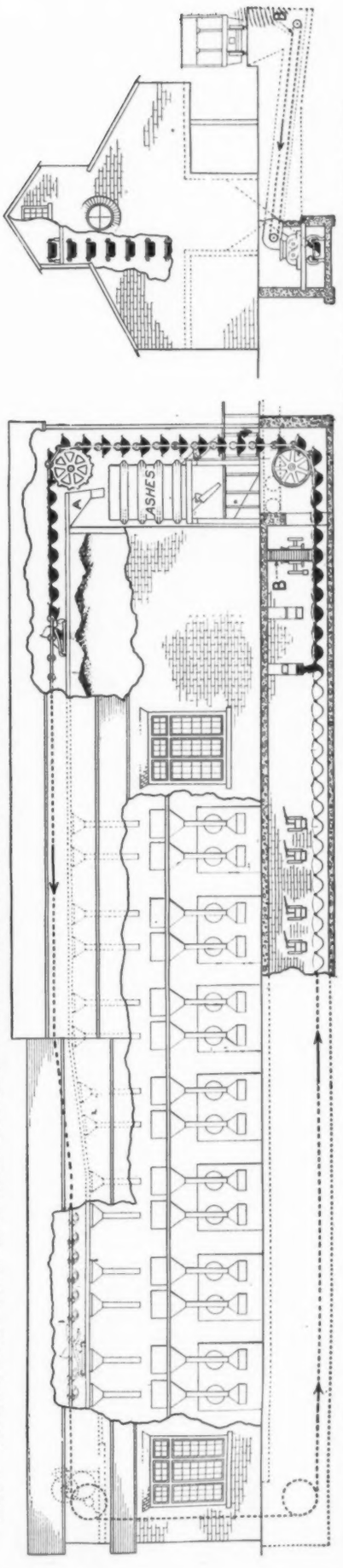


FIG. 9. SIDE AND END ELEVATIONS OF ANOTHER ARRANGEMENT OF PIVOTED-BUCKET CARRIER CONVEYOR HANDLING BOTH COAL AND ASHES

the flights of a flight conveyor and push it along in a steel trough, through which it can be discharged at the desired points by means of gates.

Fig. 3 shows an arrangement like this for handling coal from railroad cars and delivering either to a small overhead bin or to a ground-storage pile, also taking from the pile and delivering to the bin. The conveyor encircles the bin and pile, the lower run being located in a tunnel underneath the pile. When the stored coal is wanted, it is fed through gates to the lower run of the machine and taken up to the pocket. In this case there are three boiler rooms in the plant, and the coal is loaded into carts and delivered to the boiler rooms in this way.

A COMBINATION IS OFTEN EMPLOYED

Frequently in piling coal a combination of bucket elevator and flight conveyor is used, the elevator taking the coal up to the proper height and the flight conveyor distributing it along in a pile of any desired length that is not excessive for a conveyor. A combination of this kind is shown in Fig. 4. The maximum depth of the coal is about 18 ft., and the length of the conveyor is 200 ft.

Without resorting to hand-trimming of the coal—that is, simply allowing it to take its natural slope as it is discharged from the conveyor—it is possible to store about 3,000 tons of anthracite or 2,000 tons of bituminous coal, the difference being due to the difference in the natural slope of the two kinds of coal. This sort of an arrangement is quite similar to a railroad-trestle storage, but is cheaper in first cost as well as in maintenance and depreciation, unless the railroad trestle is built of permanent and therefore expensive construction.

Where the reserve storage is some distance from the boiler room, a flight conveyor in combination with a gravity-discharge machine with the reclaiming feature is sometimes used. The gravity-discharge elevator-conveyor takes the coal up and delivers it to the flight conveyor for immediate transfer to the boiler room or delivers it to the storage pile. The stored coal can be taken from storage, the same as in Fig. 3, but in this case it is delivered to the flight conveyor, which serves instead of carts to transfer the coal to the boiler room.

This kind of an arrangement is shown in Fig. 5. In this plant the amount of coal consumed per day is about 20 tons, and the amount that can be stored is about 600 tons, or 30 days' supply. Fig. 6 shows a somewhat similar arrangement. In this case the elevator is about 65 ft. from the end of the boiler room, and the space between it and the building is used for storage, being served by the flight conveyor to the boiler room.

In the opposite direction there is a flight conveyor encircling the pile and having a horizontal length of about 150 ft. When coal is being delivered to this pile, it is lifted by the elevator and carried out on the upper run of this flight conveyor. When it is being taken from the pile, it is fed to the lower run of the flight conveyor in a tunnel underneath the pile, delivered to the elevator, then taken into the boiler room by the other flight conveyor. As the firing is done by hand the coal is simply deposited on the boiler-room floor.

A PIT MAY BE USED FOR STORAGE

The arrangement shown in Fig. 7 is for small anthracite. The storage space was excavated so as to form a pit, the coal being stored in the pit and also piled up above the regular ground level. Over 11,000 tons of coal can be stored at this plant, and all except a little that is used in some boilers in an adjacent building is carted to the main boiler room of the publishing plant, a mile or so away. The machinery includes a gravity-discharge elevator-conveyor, a stocking-out flight conveyor and two reclaiming flight conveyors in tunnels under the pile. The gravity-discharge machine elevates the coal and delivers it to the wagon-loading pocket or to the pile, and also transfers coal from the pile to the pocket. The stocking-out conveyor serves to pile the coal on out in the storage area away from the pocket, and the reclaiming conveyors take it from points out of reach of the gravity-discharge machine and transfer it to this machine. The rate of handling is about 40 tons per hour.

For low maintenance cost and reliability the pivoted-bucket carrier type of conveyor excels all others for handling coal, ashes and similar materials. Some makes of these machines have been carefully perfected, so that they represent a high degree of development in the conveying art. With this type of machine the material is carried along in the buckets, which do both the elevating and distributing. The material is discharged by tilting the buckets with a tripping arrangement at the desired points, and the weight of chains, buckets and coal is supported on the horizontal runs by large well-oiled rollers traveling on T-rails. These rollers are confined between T-rails on the vertical runs, so as to guide the chain and buckets.

Two ways of using these machines in connection with reserve storage are illustrated in Figs. 8 and 9. The former has overhead-storage bins directly over the reserve storage with the carrier encircling both the overhead and ground storage, also encircling an ash pocket. When the overhead storage is depleted, the coal in the reserve storage can be fed to the lower run of the machine and transferred to the overhead bins.

With a reliable machine of this kind to transfer the coal, the overhead-storage can be made quite small and the first cost thereby reduced. Even though the replenishment of the overhead supply is dependent on the frequent operation of the carrier, there is little danger of lack of coal in the overhead bins owing to trouble with the machinery, as breakdowns with a first-class carrier are rare, and when repairs are necessary they can be made with little delay in the operation.

The other design is quite similar to this, except that the main or reserve supply is placed in a bin at one end, instead of being underneath the overhead bins. This particular carrier has a horizontal run of about 275 ft. and a vertical lift of about 42 ft., or a total length of about 630 ft. Only a little over 8 hp. is required to operate it.

Recent Court Decisions

Digested by A. L. H. STREET

Right to Ice on Mill Pond—The right of flowage belonging to the owner of a dam does not carry with it the right to take ice from the water which the dam causes to collect over the land of an adjoining owner. "The riparian owners are not, we think, deprived of their rights as such owners by the construction of a dam below them. Their right to a just and reasonable use of the water remains if it has not been actually surrendered. If any riparian owner has land covered by the pond, we see no reason why he may not take the ice from that land, unless by so doing he causes an actual injury to the owner below by materially diminishing the use of the water to which such owner is entitled." (New York Court of Appeals, Valentino vs. Schantz, 109 "Northeastern Reporter," 866.)

Nature of Power Company's Service—Generation of electricity and its transmission to consumers for power purposes does not amount to "manufacturing" or engaging in a "mercantile pursuit," within the meaning of the section of the Federal bankruptcy act which authorizes involuntary bankruptcy proceedings against a corporation engaged principally in manufacturing or mercantile pursuits, according to the holding of the United States District Court for the Middle District of Pennsylvania in proceedings brought against the Wilkes-Barre Light Co. Judge Witmer says:

> The subject of electricity is always interesting, and the study of it opens a field almost without limitation. Much has been said and written, and it yet remains shrouded in mystery, and courts remain divided as to whether electricity is a product to be manufactured. Without attempting a lengthy scientific speculation or rehearsal of conflicting views of opinions of those whose duty it has been to give this matter consideration for one reason or another, I record myself with those of opinion that generating electricity is not manufacturing, nor is the transmitting of electricity over lines of wire to consumers for a consideration a mercantile pursuit, within the meaning of the bankruptcy act.

Power Plant as Private Nuisance—Plaintiff's petition in a suit against defendant city stated a good cause of action for damages by alleging that plaintiff owned a lot within the city limits and resided upon it; that the city erected a power plant within 30 ft. of her property, for use in generating electricity and in pumping water for public use; that by reason of the noise caused in operating the plant, "its steam exhausts, and the blowing of the whistle, and by reason of the vibrations occasioned by the running of its dynamos, engines and other machinery," the value of plaintiff's home and lot had been destroyed; that the power plant was conducted by the city in such manner as to amount to a nuisance; and that without necessity it blew the whistle three times daily, at 6 o'clock morning and evening and at noon, "in long, ear-splitting blasts." (Georgia Supreme Court, Harris vs. Washington, 86 "Southeastern Reporter," 220.)

Lubricating Chain Drives is best done with thin oil first, to penetrate into the joints, followed by heavier oil, according to the working temperature and speed, applied with a brush while the chain is running slowly. Greases are not suitable for thorough lubrication of chain drives.

Higher Steam Pressures*

By Robert Cramer

SYNOPSIS—The tendency is toward the use of higher steam pressures. Tables show the increase in the ideal efficiency for various pressures and superheat. With high pressures and consequently high stack temperatures, economizers would be desirable. Large turbines would have extra heavy walls at the high-pressure ends; piping and fittings would be small for the very high pressures. A single-acting uniflow steam engine is suggested.

The history of the development of the steam engine presents a gradual but continuous and persistent rise of steam pressures, so that it seems natural that engineers should ask the question, "How far can steam pressures be practically and profitably increased?" Pleas for higher steam pressures have several times resulted from investigations of this subject; one of the most noteworthy of these was Prof. R. H. Thurston's paper on the "Promise and Potency of High-Pressure Steam," read before the society in December, 1896.

RECENT IMPROVEMENTS FAVORABLE TO HIGH PRESSURES

Professor Thurston had a clear idea of the thermodynamic principles which make possible better economy by the employment of higher steam pressures, but for several reasons he could not go as far as we today in presenting the practical aspect of the problem. Our progress in this respect can briefly be summarized by referring to the conditions which existed in Professor Thurston's time:

Superheated steam had hardly been used, and the real advantage of using it was not understood. The real cause of such cylinder condensation as is not a natural accompaniment of adiabatic expansion was not clearly understood, and it was believed that to attain a high ratio of expansion in a piston engine it was necessary to use multiple expansion cylinders. The advantages of the uniflow cycle had not been proved. Steam turbines of considerable size were in the experimental stage. The best shop methods did not permit the construction of high-pressure steam boilers without involving prohibitive expense.

Since the science of thermodynamics has been recognized by engineers as a sure guide to improvement of heat-engine economy, there has existed a tendency to increase the temperature range of the working fluid, that is, to increase the temperature at which the working medium absorbs heat and to lower the temperature at which it rejects heat. In steam-engine practice this has led to the recognition of well-defined limits: A maximum steam temperature of about 600 deg. F., above which lubrication of piston engines and maintenance of valve parts and packings are difficult, and a minimum condenser temperature of about 80 deg. F., corresponding to 29 in. of vacuum or ½ lb. back-pressure. This temperature is so near the usual cooling-water temperature that a higher vacuum would require disproportionately large quantities of circulating water. The necessary size of the circulating pump and also the necessary increase in the size of the air pump render higher degrees of vacuum unprofitable.

If, taking these limits, all heat were transferred to the steam at 600 deg. F., or 600 + 460 = 1,060 deg. abs., and all heat were rejected at 80 deg. F., or 80 + 460 = 540 deg. abs., the Carnot cycle would be realized, and the efficiency of the engine would be $\frac{1,060 - 540}{1,060} = 0.491$. Thus nearly one-half of the heat transferred to the steam would be transformed into mechanical energy.

While the condition of the rejection of all heat at the lower temperature can be closely approximated, it is impossible to cause the steam to absorb all heat at the higher temperature. In the boiler the water must be evaporated at rising temperature, and if the steam is superheated the process of superheating must also take place at rising temperature. Only that heat which converts water at a certain pressure and temperature into steam of the same pressure and temperature is absorbed at constant temperature. This heat is the larger part of the total transferred from the furnace to the water.

*A paper presented at the December, 1915, meeting of the American Society of Mechanical Engineers.

It is apparent from the foregoing that the temperature limits do not alone determine the efficiency of the process, but that the degree of approximation to the ideal Carnot cycle is of importance. In the best present practice, except for slightly higher pressures in some few isolated cases, the maximum steam pressure is 200 lb. per sq.in. absolute, and the superheat is 200 deg. F. The corresponding temperature of evaporation is 382 deg. F. and the bulk of the heat is absorbed at a temperature of 200 deg. F. below the maximum. It seems reasonable to expect that the approximation to the ideal Carnot cycle, and simultaneously the economy, would be improved by using higher pressure and less superheat; that is, by increasing the temperature at which the bulk of the heat is absorbed without increasing the maximum temperature.

Even a casual reference to steam tables and diagrams confirms this expectation and reveals the remarkable fact that the higher the steam pressure the less the total heat in the steam if the final temperature be kept constant, and correspondingly the superheat is reduced with advancing pressures. This difference, while not great, is decidedly noticeable as Table 1 shows (the values throughout are taken from the Marks and Davis steam tables).

TABLE 1. TOTAL HEAT OF STEAM AT VARIOUS PRESSURES
Temperature of steam, 600 deg. F.

Steam pressure, lb. per sq.in...	100	200	300	400	500	600	1574
Temperature of evaporation, deg. F.	328	382	417	445	467	487	600
Superheat, deg. F.	272	218	183	155	133	113	0
Total heat, B.t.u. per lb.	1323.3	1317.6	1310.7	1305.6	1301.8	1298.8	1176.0

An examination of the Mollier total-heat entropy diagram shows that the amount of heat convertible into mechanical energy in adiabatic expansion to any given back-pressure is considerably higher for high pressure and little superheat than for low pressure and much superheat, if the maximum temperature of the steam is the same in both cases. We have thus two cases making for better thermal efficiency with increasing steam pressure at constant maximum temperature—the decreasing total heat of the steam and the increasing amount of that part of the heat convertible to mechanical energy in adiabatic expansion.

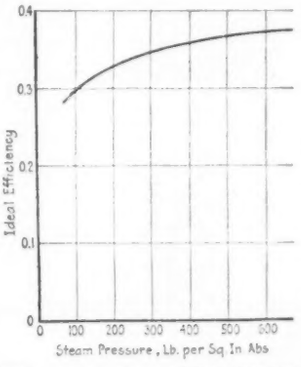


FIG. 1. IDEAL EFFICIENCY, 29 IN. VACUUM, NO SUPERHEAT

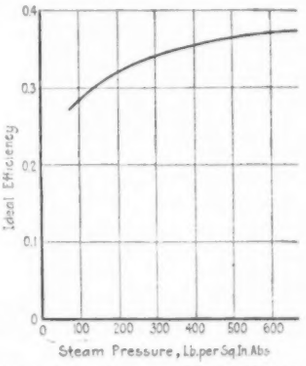


FIG. 2. IDEAL EFFICIENCY, 29 IN. VACUUM, 100 DEG. SUPERHEAT

To show clearly what gains can be expected from the increase of steam pressure, the accompanying tables have been compiled and the values contained in them plotted in the accompanying diagrams. The tables show the total heat above the feed-water temperature, which is assumed to be equal to the temperature in the condenser, and the amount of heat convertible into mechanical energy. They also show the efficiency that would be realized if the ideal Rankine cycle were carried out; that is, if the expansion of the steam were strictly adiabatic without any losses due to either heat transfer or friction, and if the expansion were continued to the condenser pressure. The tables include in addition the percentage of moisture in the steam at the end of the expansion, which will be discussed later.

Table 2 and corresponding Fig. 1 are based on the assumption that the temperature of the steam is 600 deg. F. in all cases, and that the back-pressure is ½ lb. per sq.in., corresponding to 29 in. vacuum. The steam pressure and the amount of superheat are variable quantities.

Table 3 and Fig. 2 are based on a constant condenser pressure corresponding to 29 in. vacuum and on a constant superheat of 100 deg. F., so that both steam pressure and temperature of the steam are variable. Table 4 and Fig. 3 are based on the same conditions as Table 2 and Fig. 1, except that the back-pressure is 14.7 lb. per sq.ft. corresponding to atmospheric exhaust.

Table 5 and Fig. 4 are based on the same conditions as in Table 3 and Fig. 2, but on atmospheric exhaust.

TABLE 2. EFFICIENCY AT VARIOUS STEAM PRESSURES

Temperature of steam, 600 deg. F. Back-pressure, ½ lb. per sq.in. (29 in. vacuum)

Pressure in Lb. per Sq.In. Abs.	Superheat in Deg. F.	Heat Units in 1 Lb. of Steam Converted: Total Above 80 Deg. F.	Heat Units in 1 Lb. of Steam Converted: in Adiabatic Expansion	Ideal Efficiency, per Cent.	Percentage of Moisture in Exhaust
100	272.2	1276.4	380	0.298	14.4
200	218.1	1268.6	417	0.329	18.6
300	182.5	1262.5	436	0.345	21.1
400	155.2	1257.9	454	0.361	23.2
500	132.7	1253.6	459	0.367	24.1
600	113.4	1251.0	467	0.373	25.1
1574	0	1128.0	454	0.403	35.7

TABLE 3. EFFICIENCY AT VARIOUS STEAM PRESSURES

Superheat, 100 deg. F. Back-pressure, ½ lb. per sq.in. (29 in. vacuum)

Pressure in Lb. per Sq.In. Abs.	Temperature of Steam in Deg. F.	Heat Units in 1 Lb. of Steam Converted: Total Above 80 Deg. F.	Heat Units in 1 Lb. of Steam Converted: in Adiabatic Expansion	Ideal Efficiency, per Cent.	Percentage of Moisture in Exhaust
100	427.8	1191.7	339.7	0.285	0.187
200	481.9	1209.1	387.1	0.320	0.215
300	517.5	1220.2	414.2	0.340	0.230
400	544.8	1228.9	432.9	0.353	0.240
500	567.3	1237.0	450.0	0.364	0.248
600	586.6	1244.0	461.0	0.370	0.253
684	600.0	1250.0	473.0	0.379	0.258

TABLE 4. EFFICIENCY AT VARIOUS STEAM PRESSURES

Temperature of steam, 600 deg. F. Atmospheric back-pressure, 14.7 lb. per sq.in. abs.

Pressure in Lb. per Sq.In. Abs.	Superheat in Deg. F.	Heat Units in 1 Lb. of Steam Converted: Total Above 212 Deg. F.	Heat Units in 1 Lb. of Steam Converted: in Adiabatic Expansion	Ideal Efficiency, per Cent.	Percentage of Moisture in Exhaust
100	272.2	1144	196	0.171	2.3
200	218.1	1137	224	0.197	5.9
300	182.5	1130	225	0.226	9.8
400	155.2	1126	273	0.243	12.0
500	132.7	1122	282	0.251	13.4
600	113.4	1119	291	0.260	14.7
1574	0	996	305	0.306	28.8

In all the comparisons based on constant final temperature the steam pressures are carried up to 1,574 lb., which is the pressure at which saturated steam has a temperature of 600 deg. At present such high pressures should hardly be advocated, but the figures clearly indicate the gain that might be realized with engines and turbines designed for such steam pressures.

Fig. 5 illustrates these theoretical gains due to high steam pressures and shows how the relative amount of convertible

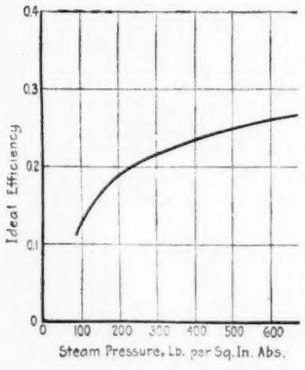


FIG. 3. IDEAL EFFICIENCY WITH ATMOSPHERIC EXHAUST, NO SUPERHEAT

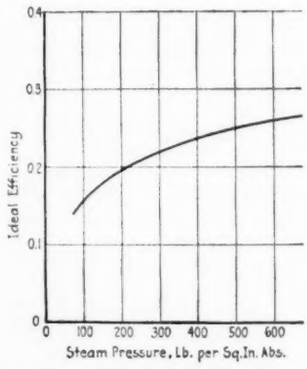


FIG. 4. IDEAL EFFICIENCY, 100 DEG. SUPERHEAT, ATMOSPHERIC EXHAUST

heat increases with higher pressures, and the tabulations in Table 6 give numerical values.

Tables 7 and 8 give the theoretical percentage of gain for certain comparisons, selected on account of their relation to conditions prevailing in present practice.

These tables show that, even in case of high vacuum in the condenser, the gains, while not overwhelmingly large, deserve careful consideration, and that in case of atmospheric exhaust these gains are so large as to fully justify an endeavor to use high steam pressures. Referring to Tables 2 and 4, we find that, theoretically, the efficiency of high-pressure noncondensing engines should be as high as that of many condensing engines under present conditions.

The following considerations show how far the benefits of high steam pressures may be realized under working conditions, as far as our present knowledge and experience enable us to realize them.

The ideal Rankine cycle cannot be realized in practice. In steam turbines the expansion of the steam cannot be effected

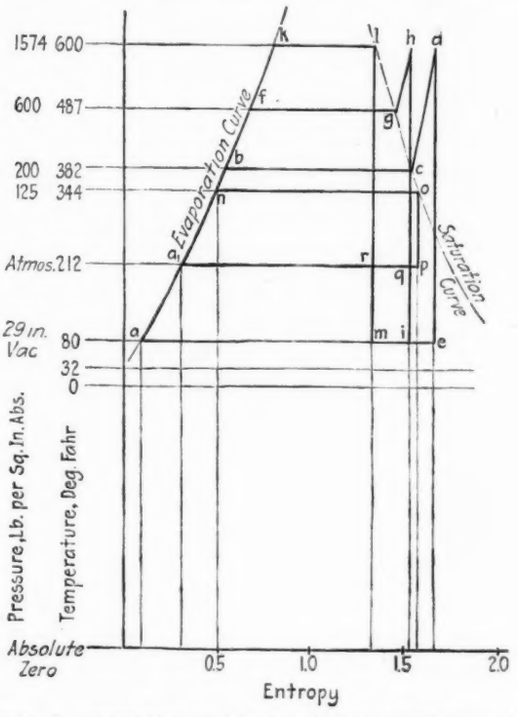


FIG. 5. ENTROPY DIAGRAM; PRESSURES UP TO 1,574 LB. ABS.

without friction losses, which are converted into heat and to that extent make the expansion deviate from the adiabatic. It is also impossible to fully extract the mechanical energy that manifests itself in the velocity of the steam, the residual velocity representing a loss.

In piston engines of the double-flow type there are large losses due to initial condensation at the time of steam admis-

TABLE 5. EFFICIENCY AT VARIOUS STEAM PRESSURES

Superheat, 100 deg. F. Atmospheric back-pressure (14.7 lb. per sq.in. abs.)

Pressure in Lb. per Sq.In. Abs.	Temperature of Steam in Deg. F.	Heat Units in 1 Lb. of Steam Converted: Total Above 212 Deg. F.	Heat Units in 1 Lb. of Steam Converted: in Adiabatic Expansion	Ideal Efficiency, per Cent.	Percentage of Moisture in Exhaust
100	427.8	1059.7	148.7	0.1405	0.062
200	481.9	1077.1	203.6	0.189	0.100
300	517.5	1088.2	235.2	0.216	0.121
400	544.8	1096.9	256.9	0.234	0.134
500	567.3	1105.0	275.0	0.249	0.144
600	586.6	1112.0	289.0	0.260	0.152
684	600.0	1123.0	306.0	0.272	0.158

sion and to heat transfer between the steam and the cylinder walls. In the uniflow type of piston engine, properly designed, these losses can be almost entirely avoided. In this latter type it is impossible to carry the expansion down to the condenser pressure, because if such were done the compression, which commences at the same point of the stroke where the expansion ends, would simply be a reversal of the expansion. Thus the compression curve would retrace the expansion curve, and no work would be done in the cylinder. In fact, no steam could enter the cylinder, because the clearance would contain as much steam of the same pressure and volume as that admitted for expansion.

These sources of loss—friction in the turbine, condensation in the double-flow piston engine and incomplete expansion in the uniflow engine—determine the practical limits of the possibility of realizing the Rankine cycle. In large steam turbines this approximation, or the Rankine-cycle efficiency, has been carried to about 76 per cent., and an efficiency

closely approximating this has been realized in uniflow engines, even in small sizes.

There is no reason why the same relative efficiency should not be realized with higher steam pressures. Such losses as are simply due to temperature differences must be the same if the initial and final temperatures of the steam remain the same, because the temperature changes through which the steam passes in performing the cycle are of the same magnitude in both cases.

In a steam turbine higher steam pressures will mean either higher velocities or more stages, both introducing higher friction losses. It is, however, to be expected that by

TABLE 6. COMPARISON OF EFFICIENCY WITH DIFFERENT PRESSURES

Exhaust, 29 In. vacuum

Initial Condition of Steam			Heat Units in 1 Lb.				
Pressure, Lb. per Sq.In. Abs.	Superheat, Deg. F.	Temperature of Steam, Deg. F.	Total Above Exhaust Temperature	Convertible in Adiabatic Expansion	Ideal Efficiency	Ratio Compared with First Line	Cycle in Temp.-Entropy Diagram, Fig. 5
200	218	600	1267	417	0.3285	1.000	a b c d e a
600	113	600	1251	467	0.3730	1.135	a f g h i a_1
1574	0	600	1128	454	0.4025	1.226	a k l m a
			Atmospheric Exhaust				
125	0	344	1010	157	0.1552	1.000	a_1 n o p a_1
600	113	600	1119	291	0.2600	1.675	a_1 f g h g a_1
1574	0	600	996	305	0.3062	1.972	a_1 k l r a_1

TABLE 7. RELATIVE GAIN IN THERMAL EFFICIENCY DUE TO INCREASING STEAM PRESSURE TO 600 LB. PER SQ.IN. ABS.

Final Condition of Steam	29 In. Vacuum		Atmospheric Exhaust	
Initial Condition of Steam	Constant Temperature, 600 Deg. F. per Cent.	Constant Superheat, 100 Deg. F. per Cent.	Constant Temperature, 600 Deg. F. per Cent.	Constant Superheat, 100 Deg. F. per Cent.
As against 100 lb. initial pressure..	25	30	52	85
As against 200 lb. initial pressure..	13	15½	32	37½

TABLE 8. RELATIVE GAIN IN THERMAL EFFICIENCY DUE TO INCREASING STEAM PRESSURE TO 1,574 LB. ABS.

Final Condition of Steam	29-In. Vacuum	Atmospheric Exhaust
As against 100 lb. initial pressure, 100 deg. superheat.....	41½ per cent.	187 per cent.
As against 200 lb. initial pressure, 218 deg. superheat.....	22½ per cent.	105 per cent.

careful design the percentage of these friction losses as compared with the total amount of energy available in adiabatic expansion can be kept the same as in present practice. In uniflow engines it is quite possible to keep the percentage of loss due to incomplete expansion as low as the corresponding percentage now obtaining.

PRESENT BOILER DESIGN AND HIGH PRESSURE

The foregoing seems to justify the employment of higher steam pressures. Standard boiler designs, however, do not permit the production of steam of a pressure higher than about 200 lb. without sacrificing safety and without calling for an investment in the boiler plant high enough to offset the gain in economy caused by higher steam pressures.

The solution of the problem of boiler safety under high steam pressures demands two fundamental changes in boiler design:

The boiler must be constructed entirely of tubes of relatively small diameter. All drums and vessels of large diameter, as well as all flat surfaces (even if stayed), must be abandoned.

Expanded, beaded or riveted joints exposed to the action of the fire must be avoided. That part of the boiler which receives the heat of the furnace must be practically a one-piece structure.

It is important that in meeting these requirements the essential characteristic of water circulation in the boiler be retained to make possible a control of the steam pressure and of the water content of the boiler by simple means. Flash boilers, while permitting high steam pressures, are not desirable, because they require complicated automatic regulating devices necessary on account of the interdependence of feed and fire control.

In considering boilers for high steam pressures, it must not be overlooked that the water in the boiler is at a higher temperature than that in a low-pressure boiler. This difference amounts to approximately 100 deg. for 600 lb. pressure, as compared with 200 lb. Even if the heating surface is made large, or, in other words, the evaporation per square foot is kept low, the stack gases will leave the high-pressure boiler at a temperature 100 deg. higher than the corresponding gases of the low-pressure boiler. The boiler efficiency, other things being equal, is correspondingly reduced.

It is possible to meet this difficulty by making the heating surface of high-pressure boilers larger in proportion to what is practiced in boilers working under pressures used today. It is also possible to increase the effectiveness of the heating surface by proper design. Boiler designers are realizing the possibilities in this direction, even with steam pressures not higher than 200 lb. This is demonstrated by modern boilers which show an evaporation per square foot of heating surface twice as high as was customary only a few years ago, at the same time realizing a better efficiency than formerly.

It seems reasonable that highly efficient high-pressure steam boilers can be made without resorting to special heat-saving apparatus to offset the effect of the higher temperature of the water in the boiler. The possibilities of these extraneous heat-saving devices should, however, not be overlooked. Even though the waste gases leaving the boiler have a temperature of 700 deg. or more, a properly installed economizer will reduce this temperature to any desired degree within the limits given by the temperature of the feed water.

The question of strength of constructional parts outside of the boiler to withstand high steam pressures is of importance. In reciprocating engines this question can be answered from practice, because pressures far exceeding 600 lb. per sq.in. are used successfully in gas engines and Diesel engines, and therefore the design of cylinders for high steam pressures should not present difficulties not already overcome.

EXTRA-HEAVY HIGH-PRESSURE CASINGS FOR TURBINES

In steam turbines, especially in those of larger size, the casings of large diameter would need to have disproportionately heavy walls were they required to withstand pressures much higher than those now employed. The solution of the problem of adapting turbines to high pressures is found in the principle underlying most present designs of large steam-turbine units. According to this principle, the pressure of the steam is greatly reduced in the first nozzle and the resulting high velocity is utilized in several rows of blades of the velocity-stage type. Thus it is possible to confine high pressures to the steam piping and to the "steam belt" carrying the steam to the first nozzles, but to have comparatively low pressure even in the first part of the casing. The lesser efficiency of the velocity staging is not so serious at the high-pressure end as it would be at the low-pressure end, because the loss manifests itself in a somewhat higher superheat of the steam entering the succeeding stages and is partly recovered.

With high pressures the difficulties with piping and fittings are reduced on account of the lower specific volume of the steam. Even if lower rates of flow than those at present customary are permitted in high-pressure steam lines (on account of the greater density of the steam), the fact that one pound of steam at 600 lb. pressure occupies approximately one-third of the space required by one pound at 200 lb. reduces the required size of the piping and fittings to such an extent that both difficulties in design and cost for a given capacity are, if anything, less than for lower pressures.

SINGLE-ACTING STEAM ENGINE SUGGESTED

In piston engines the question of cylinder lubrication is important. It is apparent that high steam pressures will permit neither slide nor Corliss valves. The advent of higher pressure will cause the poppet valve to come into its own in America, where it is now seldom used, in spite of the success it has had in Europe for many years, lately the flexible-seat type especially.

Most of the lubricating oil now used in piston engines is required for the steam-distributing valves. That quantity required by the piston and piston rings is small and the possibility of sufficient piston lubrication is not affected by higher initial steam pressures, because even in case of very high mean effective pressures in single-acting engines, where the piston serves as a crosshead and therefore requires constant lubrication, the piston can be made long enough to keep the pressure per square inch within proper limits.

For a given output the cost of a uniflow engine suitable for high steam pressures should be less than that of double-flow engines, as soon as it can be manufactured under economical conditions of manufacture. Such an engine, if single-acting, has but one simple organ of steam distribution, and on account of its high mean effective pressure its weight per horsepower is low. Even in a single-acting engine of this type the mechanism is utilized twice as efficiently as in a four-stroke cycle Diesel engine.

Steam turbines for high steam pressures would probably cost a little more than those using lower pressures and giving

the same power, on account of the extra stages required. The extra cost would be offset by a considerable saving in the required condenser cooling surface on account of the larger percentage of moisture in the steam entering the condenser and the reduced steam consumption due to better thermal efficiency. Table 9, based on the values of Table 2, will make this clear.

The question of stuffing-boxes can be eliminated in both turbines and piston engines—in turbines because high steam pressures need not be carried beyond the first nozzle, and in piston engines because the single-acting type of engine is from many points of view the logical design for high pressures. In such a piston engine the stem of the one valve can be provided with a "labyrinth" packing.

THE QUESTION OF AUXILIARIES

In very large plants all auxiliaries are often driven by electric power, and the current is furnished by a separate unit. As these "house-service units" in modern power plants have capacities of 2,000 kw. and over, it is simple to operate them directly with high-pressure steam.

TABLE 9. REDUCED STEAM CONSUMPTION DUE TO BETTER THERMAL EFFICIENCY

Initial Condition of Steam	200 Lb. 600 Deg. F.	600 Lb. 600 Deg. F.
Ideal efficiency with 29 in. vacuum.....	0.329	0.373
Percentage of steam in exhaust.....	0.814	0.749

Ratio of condenser cooling surface $= 0.329 \times \frac{0.749}{0.373} \times 0.814 = 0.812$. Thus, a saving of about 19 per cent. of the condenser cooling surface might be expected.

Where it is desired that the auxiliaries be driven independently, it should be possible to drive them with high-pressure steam turbines if the size of the auxiliary unit warrants this, or by high-pressure uniflow engines for the smaller sizes. In this latter case, the objection might be raised that the exhaust of the uniflow engine would introduce a certain amount of lubricating oil into the condenser. This can be met in different ways. If the uniflow engine is of the vertical single-acting type, the lubrication can be reduced to a small amount, and could possibly be accomplished with graphite only, so that the amount of lubricant introduced into the condenser would be insignificant and harmless.

Another way would be the noncondensing operation of the uniflow engines, using the heat of their exhaust in a feed-water heater. This manner of operation appears to be attractive, because the economy of noncondensing uniflow engines, as compared with noncondensing engines of other types, is high. Where clean feed water is available, the method would be readily applicable, or the uniflow auxiliary units may be connected to an independent small jet condenser.

In the foregoing only some of the important points in the design of prime movers utilizing high steam pressures have been touched upon, and these only in a general way. It seems clear, however, that if steam pressures are increased to, say, 600 lb. per sq.in. without using temperatures higher than those employed in modern practice, the difficulties encountered by the designer are not formidable and are more easily met than in the case of some types of explosion engines which have been successfully designed. The result to be attained by adopting such high steam pressures appears to be fully worth the effort, because thermal efficiencies closely approaching those of explosion engines can be realized with simpler and less expensive apparatus and consequently better overall economy, at the same time retaining all the practical advantages which steam utilization has over any other method of producing power.

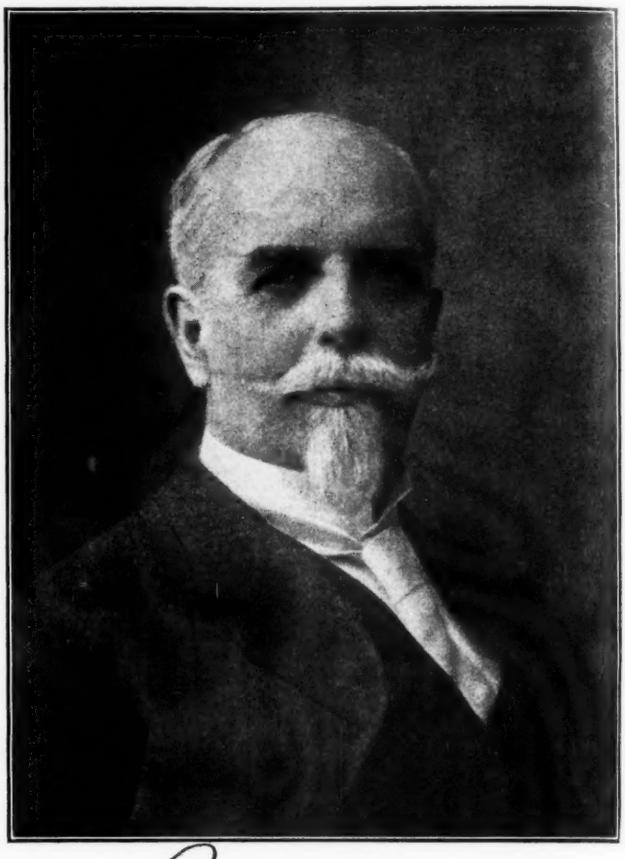
OBITUARY

JAMES MAPES DODGE

James M. Dodge, chairman of the board of directors of the Link-Belt Co., Philadelphia, Penn., died at his residence in Germantown, Philadelphia, on Saturday, Dec. 4, aged 63 years. Mr. Dodge was a native of New Jersey. His maternal grandfather, James J. Mapes, was an eminent chemist and scientist. His mother, Mary Mapes Dodge, was widely known through her editorship for many years of the juvenile magazine, "St. Nicholas." Mr. Dodge was for three years a student at Cornell University, then he spent a year at Rutgers College, New Jersey, where he took a special course in chemistry. His early engineering work was in the field of shipbuilding. Starting in as an ordinary workman at the famous old shipyards of John Roach at Chester, Penn., he became at the end of three years superintendent of erection.

In Chicago about 1880 Mr. Dodge became acquainted with William D. Ewart about the time that the latter had invented and patented a new type of sprocket-chain transmission, using separable links. Mr. Dodge joined Ewart and his associates in the commercial development of this invention, and it determined his future business career.

At the time of the Ewart invention the sprocket wheel and chain, although an old device for the transmission of motion, was hardly used in machine construction or engineering. Mr. Dodge's inventive genius found full scope in the new industry in the development of various devices intended to utilize the chain transmission. The application of chain



transmission to large power installations led to the formation in 1888 of the Link-Belt Engineering Co., where under Mr. Dodge's direction a strong engineering and scientific organization was built up.

In 1889 Mr. Dodge brought out his now well-known system of anthracite coal storage, using the Link-Belt conveyor to deliver the coal in great conical piles. A separate organization, the Dodge Coal Storage Co., was formed to exploit this invention. The extent of Mr. Dodge's original work may be judged by the fact that he took out over a hundred United States patents on his inventions.

Mr. Dodge was among the best-known members of the American Society of Mechanical Engineers. He joined the society in 1884, only four years after its organization. He was elected one of the managers of the society in 1891, was a vice-president in 1902 and was elected to the presidency in 1903. He also took great interest in the Engineers Club of Philadelphia and in the Franklin Institute. He had been vice-president of the latter organization.

BUSINESS ITEMS

The Jeffrey Manufacturing Co., Columbus, Ohio, has opened a branch office in the M. & M. Building, Milwaukee, Wis. The office will be in charge of A. Q. Dufour.

Ralph B. Phillips recently completed a merger of munition plants, and a company known as the Phillips Corporation is to be incorporated in Delaware with an initial issue of $25,000,000 cumulative, preferred and common stock. This concern is to be a holding company for the American Steam